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DESIGN OF RECIPROCATING SINGLE CYLINDER EXPANDERS FOR RANKINE CYCLE ENGINES

General Electric Company

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QUARTERLY PROGRESS REPORT

DESIGN OF RECIPROCATING SINGLE CYLINDER EXPANDERS FOR RANKINE CYCLE ENGINES

ENVIRONMENTAL PROTECTION AGENCY
CONTRACT NO. 68-01-0408
PROGRAM MANAGER–R. D. BROOKS

January 4, 1972 to April 4, 1972

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Details of illustrations in this document may be better studied on microfiche.

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1.0 SUMMARY

The Environmental Protection Agency is sponsoring development programs to identify automobile engines having low exhaust emissions. One approach being investigated is the Rankine cycle which includes a reciprocating steam expander as the prime mover. For maximum engine performance, high temperatures and pressures similar to modern steam power plant practice are required. For these operating conditions, liquid lubricants for cylinder and inlet valve lubrication are a limitation. Solid lubricants, to be investigated in this program, appear to offer long life at high temperatures. Potential system contamination problems from liquid lubricants introduced to the working fluid are also reduced.

Phase I of this program includes the design of practical single cylinder reciprocating expanders for the purpose of evaluating solid lubricants and other supporting materials for use in a Rankine cycle automotive propulsion system utilizing a water base fluid. The Phase I investigation has the following objectives:

- Establish preliminary designs of trunk piston and crosshead piston single cylinder expanders.
- Evaluate expander materials technology for higher temperature operation with solid lubricants.
- Evaluate solid and liquid lubricants for reciprocating expanders.
- Build and test one trunk piston expander and one crosshead piston expander at conditions representative of the engine system.

Phase I offers the opportunity to conduct studies of expander design and materials trade-offs utilizing selected advanced technology that will

be appropriate for the automotive application. The program is comprised of a four task effort arranged and scheduled to adequately compare the two expander types using similar materials of construction. The efforts of the second quarter are summarized in this report and include the following results:

1. Additional design analyses for lubricant wear which indicate the life for the graphite rings should be satisfactory. Stress analyses were completed with acceptable factors of safety for all locations examined. A dynamic analysis of the recompression valve established its operation to be acceptable except for very high condensor pressures (Pc = 100 psia).

2. Planning was initiated for assembly of cost data on primary and alternate materials for both the crosshead piston and trunk piston designs.

3. The Mobil synthetic hydrocarbon oil was selected for use in the first single cylinder test. It appears to have about 100°F advantage in use temperature over petroleum base stocks.

4. All parts for the crosshead piston expander were received and the test unit buildup was completed. After installation in the test cell a short motoring test was conducted. Evaluation of the test results required several modifications to the expander and test facility.

5. All parts are on order for the trunk piston expander and it is expected to be ready for test by the end of April.

6. Evaluation of the tappet and cam damage during the motoring established that the tappet material lacked adequate hardness for satisfactory operation. It is to be replaced by a chilled case iron tappet contributed to the program by Eaton Manufacturing Company.

2.0 TASK I EXPANDER DESIGN

2.1 Expander Performance Analysis

2.1.1 Trunk Expander Piston Rider Ring Wear Prediction

The basic approach followed for predicting the wear rate of the graphite rider ring for the trunk piston is presented in References (1) and (2). The analysis assumes certain ideal conditions which are not expected to be present in the actual single cylinder expander. For example the analysis (1) makes no provisions for wear due to foreign abrasive particles, (2) makes no allowances for wear due to ring misalignment or rocking in the piston ring groove, and (3) makes no allowances for wear caused by rough surfaces such as scored cylinder walls.

In the rider ring wear analysis, it is assumed that there are no gas (steam) pressure forces acting on the rider ring since the ring is notched on its rubbing face so that it does not become a pressure sealing ring. Therefore, the only load on the rider ring is the resultant force imposed by steam pressure on top of the piston and the opposing force from the connecting rod. Figure 2.1-1 shows the resultant total side forces on the trunk piston as a function of crank angle. The trunk piston is designed so that the graphite rider ring supports only one half of this load. The bottom skirt of the piston supports the other half of the piston total side load.

The cyclic average of all positive side loads on the graphite rider ring is 210 lbs., and the cyclic average of all negative side loads (on the opposite side of the piston) is only 20 lbs. Only the 210 lb. load will be considered in the following discussion. The width (axial length) of the graphite rider ring to be considered is 0.7 inches.

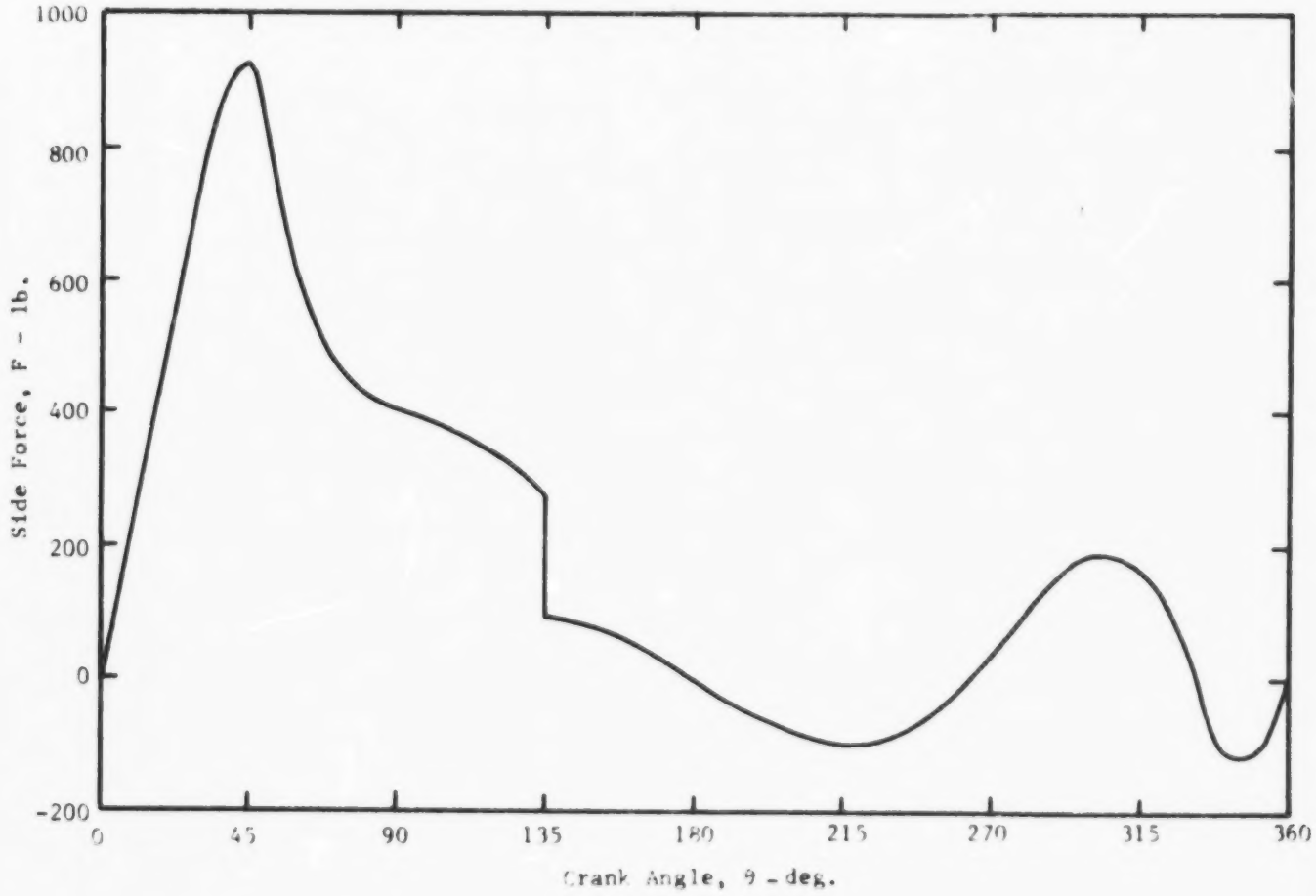

Figure 2.1-1. Wrist Pin Side Force, Due to Pressure Force, Reciprocating Weight Inertia, Force, at 2500 RPM, and Inertia Force Due to Angular Acceleration of Connecting Rod-Trunk Expander.

For a ring outside diameter which is less than the cylinder diameter, the Hertz formula (Reference 3) for the contact stress is:

$$\text{Max } S_c = q_o = 0.798\sqrt{\frac{[P]\,\frac{D_1 - D_2}{D_1 D_2}}{\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]}}, \text{ psi} \qquad (1)$$

where D_1 = Cylinder dia. = 3.5 inches

D_2 = Ring dia., inches

$E_1 = 3.5 \times 10^6$ lb/in^2 (graphite)

$E_2 = 30 \times 10^6$ lb/in^2 (steel)

$\nu_1 = \nu_2$ = Poisson's ratio = 0.3

P = load/linear inch = 210 lb/0.7 inch

Equation (1) reduces to:

$$q_o = 13711\sqrt{\frac{3.5}{D_2} - 1}\ , \text{ psi} \qquad (2)$$

For conforming geometries, the maximum shear stress, τ_{max}, is a function of q_o, a corner or edge stress concentration factor K, and the coefficient of friction μ. (Reference 1).

$$\tau_{max} = Kq_o\sqrt{0.25 + \mu^2} \leq \left(\frac{2000}{N}\right)^{1/9} \gamma_R\, \tau_y\ , \text{ psi} \qquad (3)$$

The latter half of the above expression are the conditions required to ensure zero wear for a life of N passes for a given mechanism with the fractional parameter $\gamma_R = 0.20$ (Reference 2) and shear yield strength of 7500 psi.

Solutions are shown in Figure 2.1-2 for the graphite wear rings for expander rotative speeds of 500 - 3000 RPM, for coefficients of friction $0.05 \leq \mu \leq 0.50$, and for diametral differences $(D_1 - D_2) - \Delta D$ of 0.1, 1, and 10 mils.

The fractional area of the rider ring with a stress concentration factor less than the allowable for zero wear in 1000 hours was determined and plotted in Figure 2.1-2.

It may be concluded that the higher contact stresses generated by a large difference in ring and cylinder diameters will cause greater-than-zero wear over a significant fraction of the ring face, while only very small regions near the edges of the ring will have greater-than-zero wear when the diameter difference is small. At least two conditions will tend to reduce an initially large diameter difference: (1) mechanical or thermal loads could spring the ring ends outward and increase the effective diameter of the ring, and (2) the ring will wear toward exactly the same diameter as the cylinder with time. Either should decrease the initial wear rate significantly. Also, as the ring edges wear from sharp corners to circular arcs, stress concentration factors and wear rates should decrease.

Using calculated operating temperatures and manufacturing drawing tolerances, the average ΔD is 11 mils for the 440C stainless steel cylinder liner and graphite rider ring. Initially, a large wear rate can be expected to occur, but the wear rate should decrease significantly as the wear process laps the ring and cylinder toward the same diameter and the ring edges toward circular arcs.

A satisfactory wear life is expected, based upon the simplified and ideal model developed.

2.1.2 Stress Analysis - Trunk Piston Expander

The piston position, velocity, and acceleration; cylinder swept volume; and inertia force are shown on Figure 2.1-3 as functions of crank angle for a rotative speed of 1000 RPM and for the 5.954 lb. reciprocating weight which acts on the wrist pin.

The pressure forces on the piston were calculated using a cycle with 15% cut off ($\sim 40°$ ATDC) and isentropic expansion according to:

$$PV^{1.28} = 1000\ (6.975)^{1.28} = 12016 = \text{constant}$$

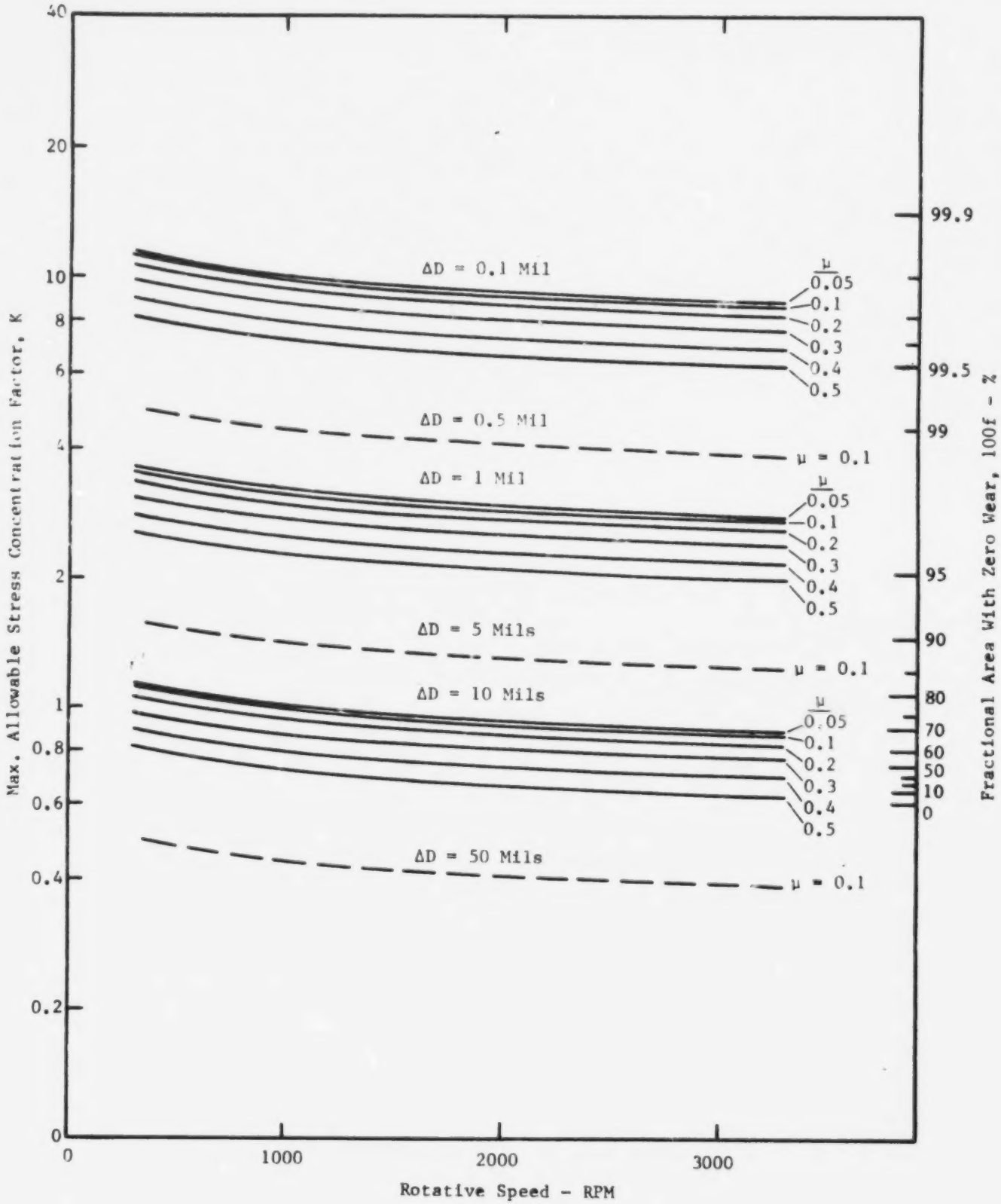

Figure 2.1-2. Maximum Allowable Stress Concentration Factor and Fractional Area With Zero Wear of Graphite Trunk Piston Rider Ring in 1000 Hours.

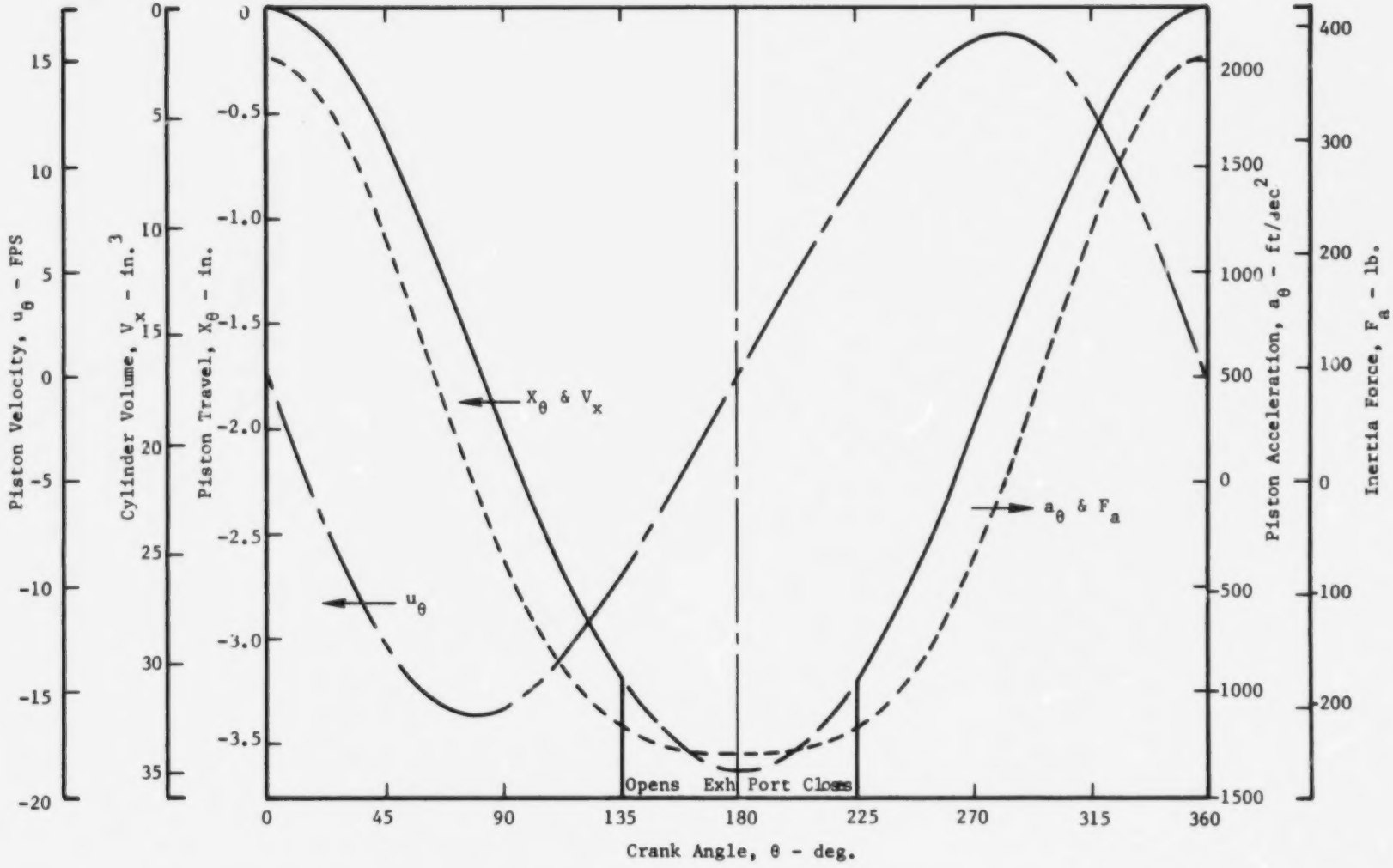

Figure 2.1-3. Piston Position, Velocity, and Acceleration; Cylinder Swept Volume; and Inertia, Force for 1000 RPM and 5.954 lb. (Reciprocating Weight Acting on Wrist Pin Only.) - Trunk Expander.

from 40° to exhaust port opening at $\theta = 135°$. The pressure remains at the condenser pressure of 20 psia until exhaust port closing at 225°. Isentropic compression occurs from 225° to 360° according to:

$$PV^{1.28} = 20\ (32.540)^{1.28} = 1726 = \text{constant}.$$

This model pressure cycle, shown in Figure 2.1-4, results in a mean pressure of 290 psi and a mean temperature of 575°F during the cycle.

The vertical pressure and inertia forces on the wrist pin are shown on Figure 2.1-5 as a function of crank angle. By combining the two forces, the total vertical force on the wrist pin under conditions of zero inlet pressure (motoring) or for 1000 psia inlet pressure for rotative speeds of 2000 and 2500 RPM can be determined.

The pressure and inertia forces at 2500 RPM on the wrist pin are combined and converted into a side force as a function of the crank angle in Figure 2.1-1. It also represents the side load on the trunk piston rider ring.

Table 2-1 and Figure 2.1-6 identify the stresses and factors of safety at the principal points of interest in the trunk expander.

One area of possible high stress in the piston head is at location A, the bottom of the bolt head hole. However, after combining the compressive and shear stresses into an effective stress and using the allowable alternating stress of the material for 1000 hr. at 600°F, the factor of safety (FS) is 9.90 against fatigue failure. This is satisfactory even if a relatively high stress concentration factor (SCF) exists.

The junction of the crown and walls of the piston (Loc. B) is difficult to analyze exactly with standard formulas because of the rapid changes in cross section from crown to groove to land, but very generous FS exist even assuming that the inlet pressure leaks past the top ring.

The land bending FS = 12.9 (Loc. C), and the land deflection is only 0.17 mils for the piston using the graphite rings. The piston using the Inconel X-750 rings is even safer. With a minimum clearance of 0.003 in. between the lands and the rings, no binding of the piston rings will occur.

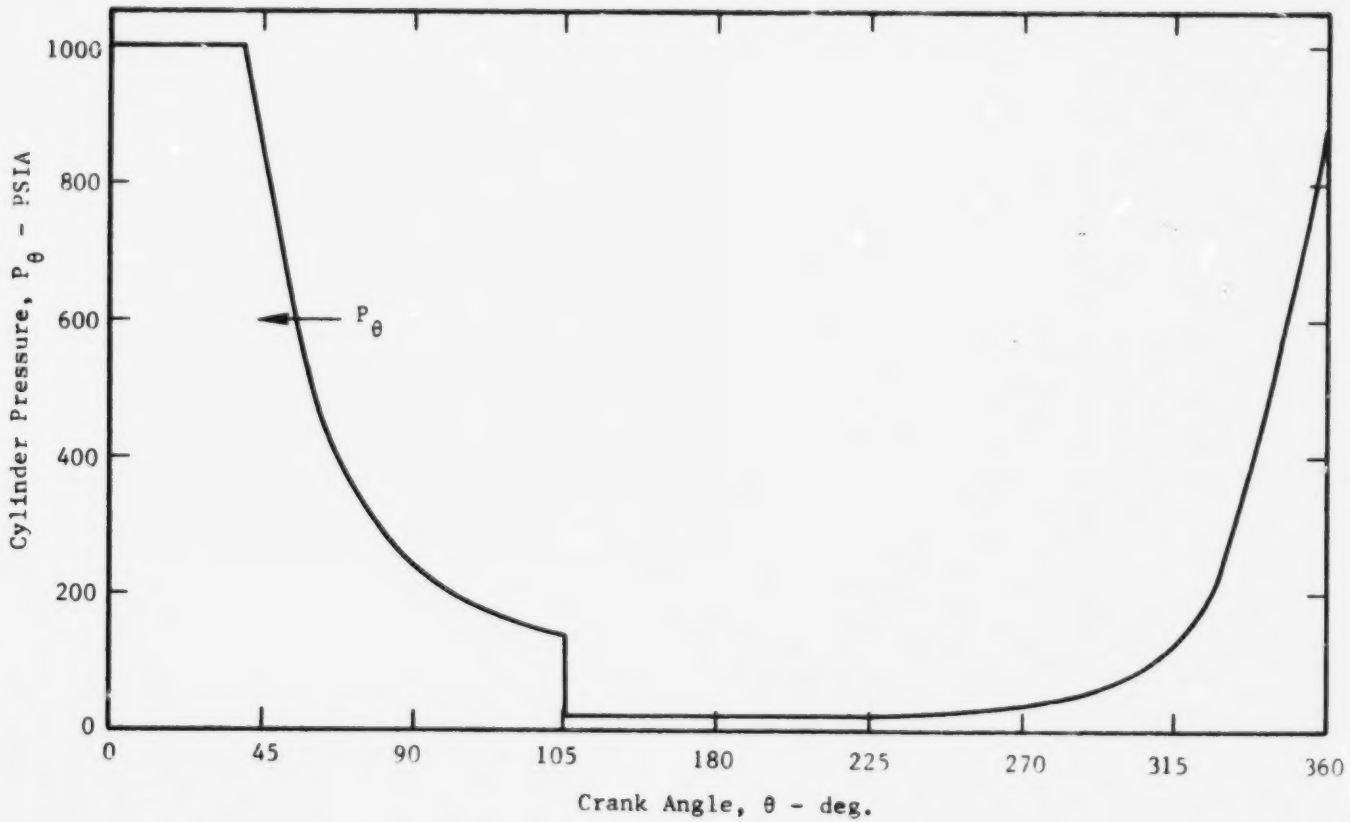

Figure 2.1-4. Model Cylinder Pressure Cycle - Trunk Expander.

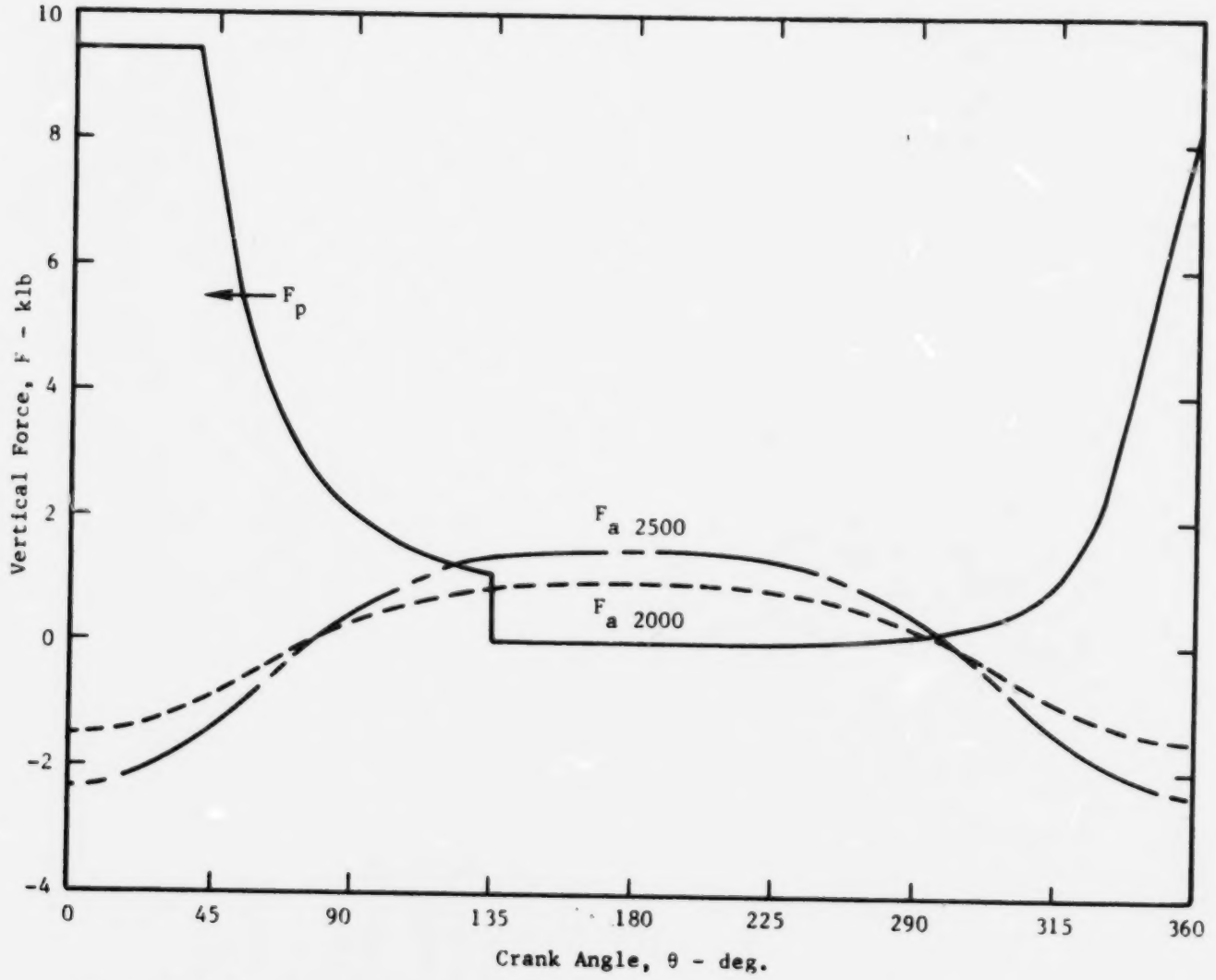

Figure 2.1-5. Vertical Wrist Pin Forces - Trunk Expander. F_p = Pressure Force. F_a = Inertia Forces at 2000 and 2500 RPM.

Table 2-1

Stresses and Factors of Safety at Principal Points of Interest in the Trunk Expander

(Fig. 2.1-6 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable[a] Stress, ksi	Factor of Safety, -
A	70.2 ± 3.2	± 31.5	9.90
B	10.0 ± 10.0	± 47.5	4.75
C	3.7 ± 3.7	± 47.5	12.9
G	4.9	11.6	2.38
H	4.6 ± 2.7	± 5.0	1.86
I	51.3 ± 22.3	± 47.0	2.10
J	2.0 ± 2.0	± 6.7	3.36
K	34.3 ± 23.5	± 61.8	2.63
L	82.5 ± 6.2	± 33.5	5.41
M	17.6	39.6	2.25
N	42.4 ± 42.4	± 75.0	1.77
O	17.7 ± 17.7	± 42.0	2.37
P	78.0 - 88.1	125.0	1.42[b]
Q	4.2 ± 3.8	± 46.6	12.2
R	71.0 ± 12.2	± 39.3	3.21
S	36.8 ± 36.8	± 44.3	1.20
T	2.9 ± 2.9	± 29.6	10.1
U	4.5 ± 4.5	± 44.3	9.88
V	14.3 ± 14.3	± 31.0	2.17
W	65.6 ± 3.9	± 41.8	10.7
X	67.8	81.0	1.19
Y	1.1 ± 1.0	± 18.5	17.1
Z	30.7 ± 27.3	± 44.8	1.64
AA	41.6	84.0	2.02
AB	43.9	120.0	2.82
AC	27.0	32.0	1.19
AD	5.5 ± 5.5	± 16.0	2.94

Table 2-1 (Cont'd.)

Stresses and Factors of Safety at Principal Points of Interest in the Trunk Expander

(Fig. 2.1-6 Identifies Location)

Location	Effective Stress Cycle, ksi	Allowable[(a)] Stress, ksi	Factor of Safety, -
AE	12.3	32.0	2.60
AF	2460 lb	3590 lb	1.46[(c)]
AG	10.8 ± 10.8	± 20.0	1.86
AI	77.2 ± 16.8	± 36.0	2.14
AJ	15.1 ± 15.1	± 20.0	1.33

NOTES: (a) The allowable stresses shown as ± are the allowable alternating stresses when the mean stress is the same as shown in the effective stress tabulation.

(b) Spring - maximum allowable stress is 125.0 ksi for initial stress of 78.0 ksi. Maximum imposed stress is 88.1 ksi.

(c) FS = critical load/applied load.

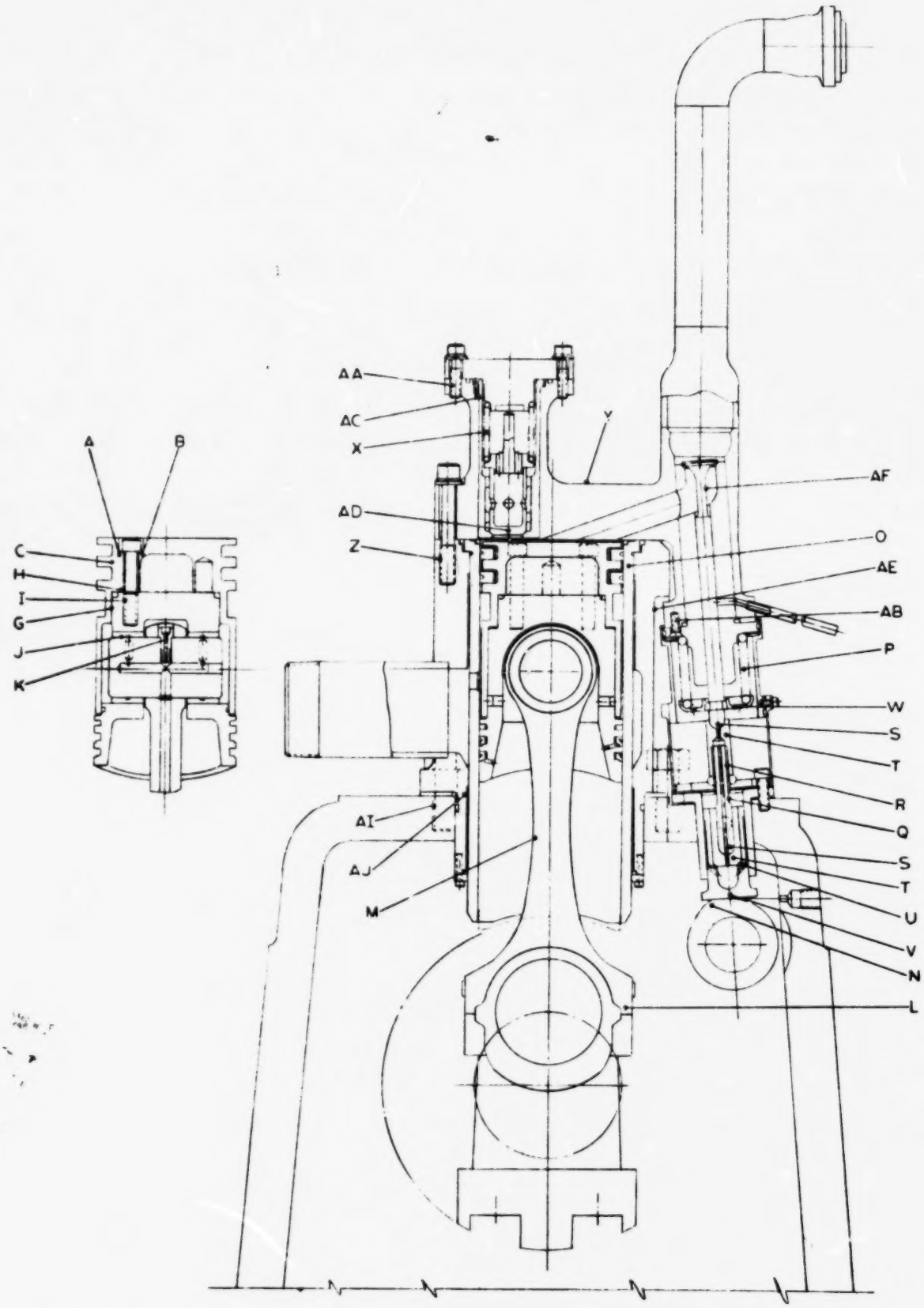

Figure 2.1-6. Critical Stress Locations for Trunk Expander (Refer to Table 2-1). (Dwg. 707E729).

The difference in thermal expansion coefficients between the aluminum skirt and the steel crown can cause high compressive stresses in the aluminum and high tensile stresses in the steel at elevated temperatures. With dimensions set to guarantee 0.004 in. diametral clearance at assembly (Loc. G), FS = 2.38 in the skirt and 2.44 in the thin crown wall inside the rider ring groove, with both parts at the calculated operating temperature of 515°F.

The alternating bearing stresses between the piston head and the aluminum skirt (Loc. H) are satisfactory, with FS = 1.86.

The most critical problem for the piston crown screw (Loc. I) is that of motoring at high speed. During motoring, the bolts must absorb the crown inertia forces, while the pressure forces during power generation would tend to unload them. With a specified bolt preload, and with a force of 2260 lb. due to differential thermal expansion between the aluminum and the steel, and using a SCF = 3.85 for thread fatigue, the thread FS is 2.10 for motoring at 2500 RPM for 1000 hrs. This FS is against the bolt breaking. Considering the endurance strength of the aluminum skirt into which the bolts are threaded and the thread engagement length, the FS = 1.31 that the bolts will break before the aluminum threads strip, giving FS = 2.75 against the threads stripping. A temperature of 300°F was assumed during motoring for the above calculations.

Because of geometrical limitations, it was not considered feasible to make the piston crown which uses Inconel X-750 rings as light as the one using graphite rings. Two different reciprocating weights must be balanced. This requirement was met by designing the counterbalance weights with two removable plugs.

The projected wrist pin pressures (Loc. J) are compressive in one direction during the power stroke and compressive in both vertical directions during motoring. Neither exceed 4000 psi and are acceptable. The stresses due to the inertia loads of the piston, skirt beneath the wrist pin are negligible.

For 1000 hr. fatigue loading at 300°F, FS = 2.63 for the wrist pin with SCF = 3 included because of the oil hole at the maximum stress point (Loc. K).

The connecting rod bolts (Loc. L) have FS = 5.41 for 1000 hr. motoring operation at 300°F, with a fatigue SCF = 3.85 assumed.

Against short-column buckling, the connecting rod FS = 2.25 (Loc. M). Against alternating axial and bending stresses including the whipping effect the FS = 3.49.

The original manufacturer's crankshaft design is being used, and it is understood that the crankshaft design has withstood pressure loads and speeds similar to those planned for significant periods of time. No crankshaft stress analyses were made.

The maximum bending stresses in the camshaft is less than 800 psi, unbalance loads included, and is satisfactory. The cam Hertzian stress FS = 1.77 vs. fatigue failure (Loc. N).

The region of maximum interest in the cylinder block liner is at Loc. O where the cylinder support changes from complete restraint by the block to completely unrestrained. One liner will be made of 440C stainless steel; its replacement of 1Cr-1/2 Mo-1/4 V steel. The liners will be shrunk fit at assembly so that line contact still exists at the calculated 540°F operating temperature. For 440C the assembly FS = 5.12, and for 1Cr-1/2 Mo-1/4 V the assembly FS = 78 against room temperature yielding.

During operating, the combined effect of pressure forces, an assumed $\Delta T = 200°F$ across the cylinder wall, and the change in restraint produces FS = 4.57 in the 440C liner and FS = 2.37 in the 1 Cr-1/2 Mo-1/4 V liner against 1000 hr. fatigue at 600°F.

A compression spring (Loc. P) of standard size and material was chosen which provided a preload of 177 lb. and a maximum force of 200 lb. at 0.100 in. cam lift. The corresponding alternating stresses of 78.0 - 88.1 ksi provided FS = 1.42 against fatigue. Its natural frequency of 17,095 cpm was satisfactory for the first six harmonics of a 2500 RPM driving force. Buckling is not a problem.

All of the valve train components (Loc. W) were analyzed with respect to fatigue and had FS = 1.20 - 12.2. The push rod FS = 4.81 against buckling.

The cam profile prevents impact loading of the inlet valve sealing faces.

A standard size compression spring (Loc. X) was chosen for loading the recompression valve. Inconel X-750 material was specified because the spring may be continually soaked in the 900°F inlet steam. After an assumed three percent relaxation, the spring preload will be 78 lb. Since this valve is designed to operate only accasionally, fatigue is not a problem and the corresponding stress of 67.8 ksi produces a FS = 1.19 vs. the proportional limit of the material at 900°F. The first critical frequency of 18,150 cpm was satisfactory for the first seven harmonics of 2500 RPM. Buckling is not a problem.

The selected preload of the recompression valve spring sets the valve life-off pressure at 214 psia during motoring at ambient pressure, and 1200 psia when the inlet pressure is 1000 psia. The 1200 psia opening pressure is 447 psi less than that required to open the steam inlet valve at 1000 psia inlet pressure. The 214 psia opening pressure for the recompression valve is 193 psi less than that required to open the inlet valve at 14.7 psia inlet pressure. (Recompression valve dynamic analysis is presented below.)

The inlet manifold head FS = 17.1 (Loc. Y) without consideration of the SCF of holes, ports, and instrumentation fittings. None of these are expected to produce unacceptable conditions.

The head bolts (Loc. Z) and all others in the expanders are provided with a preload which gives FS $\geq$ 1.25 against the opening of their joints by pressure forces. The head bolt FS = 1.64 against fatigue, with SCF = 3.85 assumed, for 14.4 ft-lb assembly torque. For a greater assembly torque, such as 40 ft-lb, the fatigue FS = 0.88 vs. lower 3σ strength properties but FS = 1.36 vs. average strength properties at 500°F.

The recompression valve and inlet valve housing bolts are subjected only to steady loads. The recompression valve bolt (Loc. AA) FS = 2.02 vs. yielding at 900°F and SCF = 1. Some yielding can occur if the same bolts are used for 1000 hr., since the applied stress exceeds the allowable 1000 hr. creep strength of the bolt material at 900°F.

The inlet valve body screw (Loc. AB) FS = 2.82 against yielding at 300°F with SCF = 3.

The recompression valve body (Loc. AC) has a minimum FS = 1.19 against rupture in 1000 hr. at 900°F, without consideration of the generous fillet radius at this location which will increase its safety.

The recompression valve (Loc. AD) FS = 2.94 against 1000 hours fatigue at 900°F.

The upper cylinder block (Loc. AE) FS = 2.60 against 1000 hour rupture, assuming full axial loads and 1000 psia inside.

The intake valve stem (Loc. AF) FS = 1.46 against buckling.

The base bolts (Loc. AI) FS = 2.14 against 1000-hour fatigue with SCF = 3 included.

The lower cylinder/flange intersection of the lower cylinder block (Loc. AJ) FS = 1.33 against 1000-hour fatigue.

Two counter rotating shafts with counterweights are included in the Waukesha crankcase to balance the primary (1/rev) component of the inertia force due to the reciprocating parts of the engine. A secondary force component (2/rev), which is smaller, will not be balanced. The reciprocating masses include the power piston crown and skirt, 1/3 of the weight of the connecting rod shank, and the upper end of the connecting rod with wrist pin and bearings. These weights are tabulated in Table 2-2. The actual measured weights of the crosshead expander components are tabulated, and are 98.2% of the previously reported calculated total weight.

The existing counterweights in the Waukesha crankcase do not contain sufficient mass, and geometrical limitations prevent the addition of larger size masses to the existing counterweights. However, high density sintered material with high tungsten content (Mallory 1000) will provide the proper balancing mass within the crankcase geometrical limitations. Stress levels in the counterweights and their bolts are 2.3 ksi and 46.2 ksi at 2500 RPM. Their respective factors of safety are 32 and 2.88 against fracture at 300°F.

Table 2-2

Reciprocating and Rotating Weights

	Reciprocating	
	Crosshead Expander*	Trunk Expander
Power piston head	3.043 lb.	5.683 lb.
Power piston rod	1.283	-
Crosshead piston	1.495	-
Crosshead bushing (2)	0.261	0.271
Wrist pin	1.281	1.359
Connecting rod	0.743	0.780
Total	8.106 lb.	8.093 lb.

	Rotating	
	Crosshead Expander*	Trunk Expander
Connecting rod	3.216 lb.	3.319 lb.
Connecting rod bearing	0.233	0.254
Total	3.449 lb.	3.574 lb.

*Actual (measured) weights.

The crankshaft counterweights balance the expander rotating components. Since the Waukesha crankshaft is already balanced for a specific rotating weight, the existing balancing weight need only be adjusted to compensate for the difference between the rotating mass of the Waukesha engine and that of the steam expander. The existing bolts are to be used, with small amounts of additional weight added beneath their heads. The rotating weights for the two expanders, excluding the crankshaft, are listed in Table 2-2. The measured weights of the crosshead expander rotating components are tabulated, and are 98.8% of the previously reported calculated total weight. Similar accuracy is expected for the trunk expander weights.

2.1.3 Recompression Valve Dynamics

A finite time increment analysis was made of the recompression valve dynamics. A constant increment of one crank-angle degree was used which gave time increments of 1/3000 sec at 500 RPM to 1/15000 sec at 2500 RPM. The computation method was as follows:

- Assume a condenser pressure, and inlet pressure = 1000 psia.
- Start calculations at the crank angle at which the cylinder pressure produced $\Sigma F > 0$ on the recompression valve. For $PV^{1.28}$ = constant, the recompression pressure P_{re} = 1200 psia for a 78 lb. recompression spring preload.
- Determine recompression valve acceleration from F = ma, or a = F/1618.76 lbm sec^2/in.
- Determine valve initial, average, and final velocities upward for each time period.
- Determine valve initial, average, and final lifts upward for each time period.
- Determine average valve flow area for each time period.
- Determine average valve flow coefficient for each time period.
- Determine average steam flow through valve for each time period.
- Deduct steam flow through valve from cylinder volume at the beginning of the next time period and repeat calculations for the new time period.

- Continue calculations until the inlet valve opened at either -20° or -10° by cam action (acceptable solution) or until the recompression pressure reached 1670 psia and produced $\Sigma F>0$ on the inlet valve (unacceptable solution).

Samples of such calculations are shown on Figure 2.1-7. All of the conditions shown except for 1000 RPM, P_c = 100 psia, are acceptable since the inlet valve will open from cam action before 1670 psia recompression pressure is reached.

The results of all calculations are summarized on Figure 2.1-8, which shows the allowable variation of condenser pressure with RPM. Less favorable results occur when the inlet pressure is less than 1000 psia.

2.1.4 Crosshead Expander Power Piston Natural Frequency

The transverse bending natural frequency of the crosshead power piston and its rod as a cantilevered beam with the piston mass at its free end was calculated using the formula (Reference 4):

$$\omega^2 = \frac{k}{M+0.23\ m} = \frac{3EI}{L^3\ (M+0.23\ m)} \quad rad^2/sec^2 \qquad (4)$$

where: $E = 28 \times 10^6$ psi (rod at 300°F)

$I = 0.01508\ in^4$ (rod)

$L = 5.75$ in.

$mg = 0.6217$ lb (rod-calculated)

$Mg = 3.043$ lb (piston-measured and calculated)

Then $\omega = 899$ rad/sec, and $f = (\frac{\omega}{2\pi})\ 60 = 8580$ cpm.

Therefore, the rod/piston system is satisfactory for the first three harmonics of the maximum operating speed of 2500 RPM, and for higher harmonics for lower operating speeds.

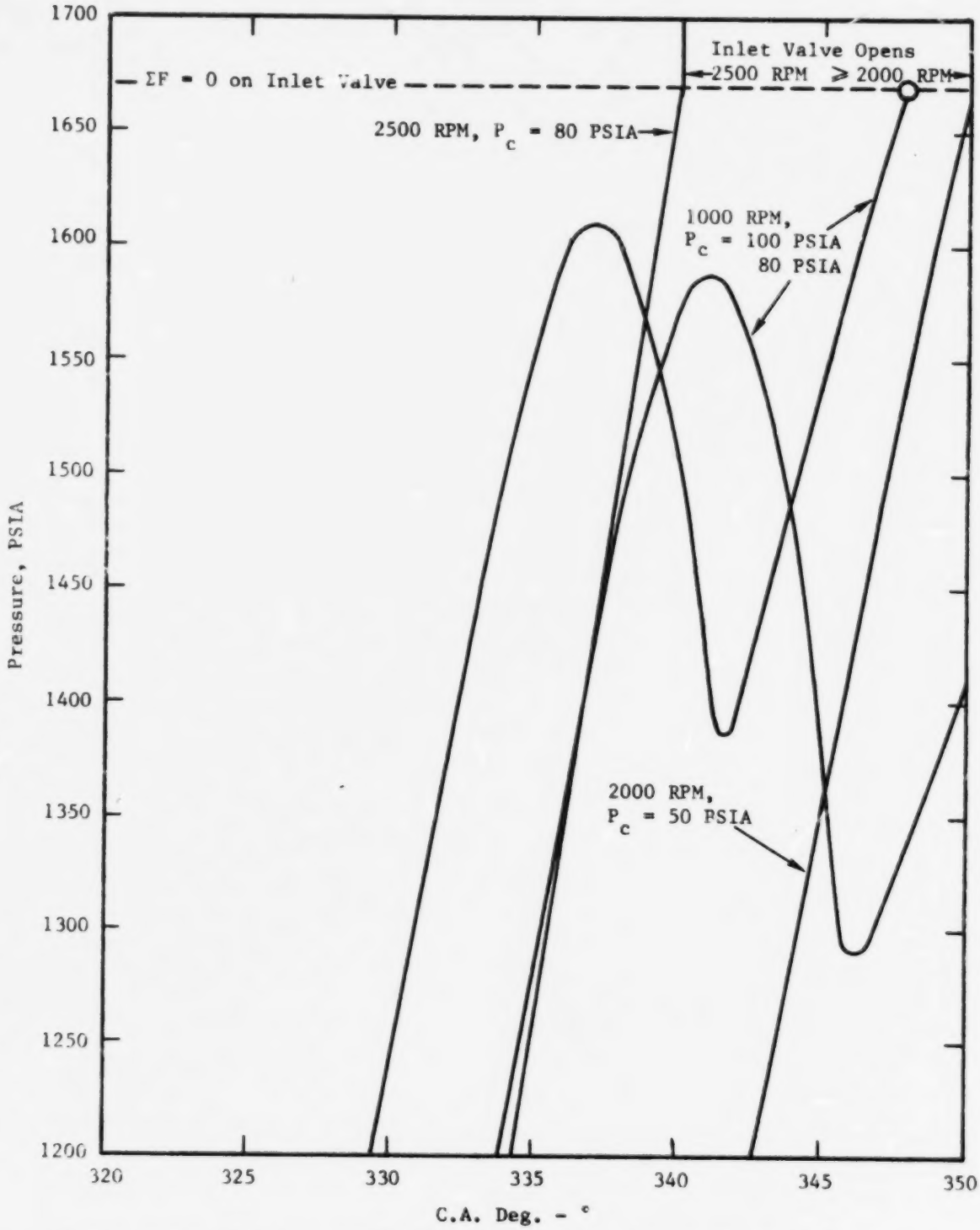

Figure 2.1-7. Examples of Recompression Valve Dynamic Calculations for Inlet Pressure of 1000 PSIA and Different Rotative Speeds and Condenser Pressure, P_c. All Solutions Shown are Acceptable Except for 1000 RPM, Where P_c = 100 PSIA.

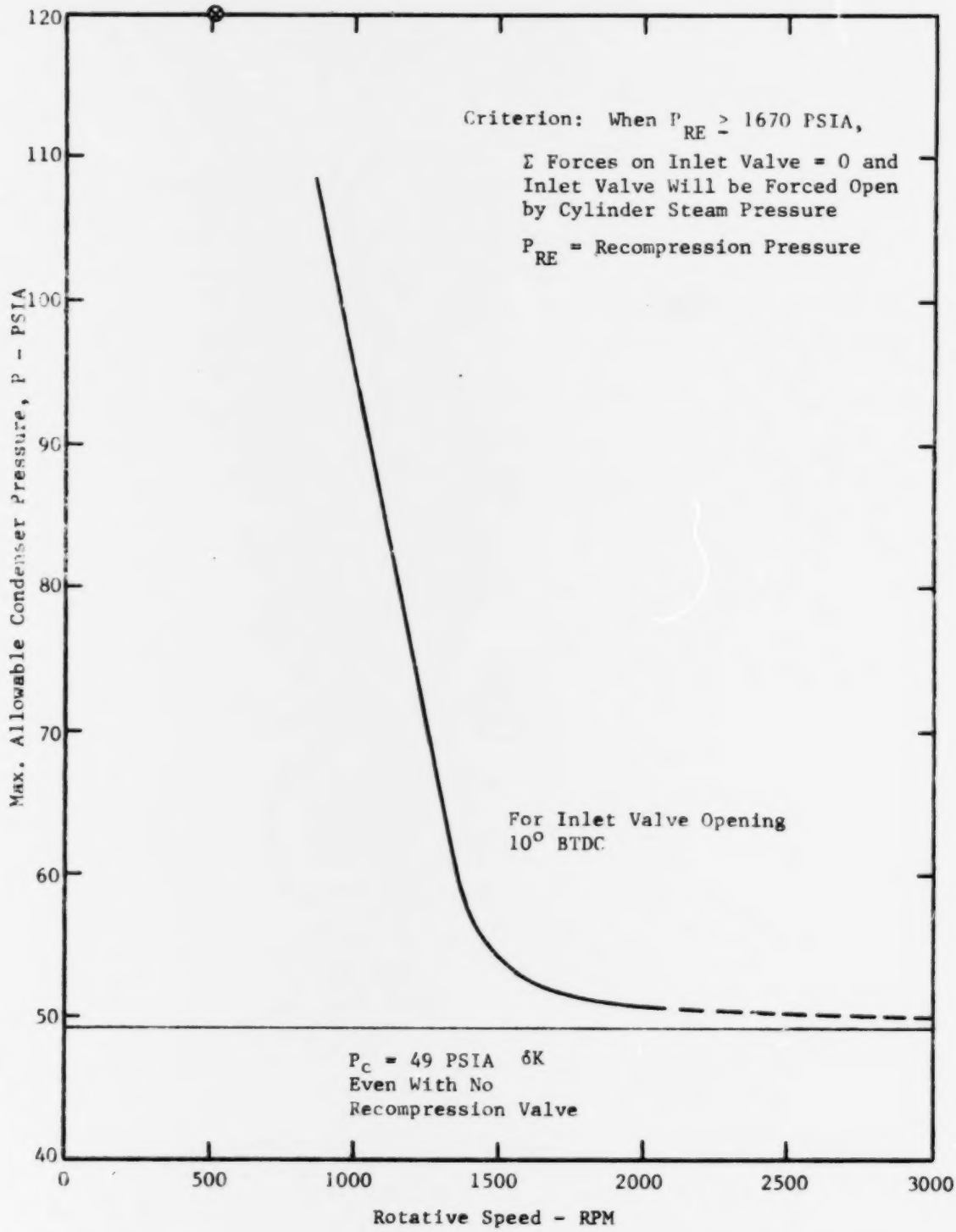

Figure 2.1-8. Maximum Allowable Condenser Pressure for Inlet Pressure of 1000 PSIA and 78 lb. Recompression Valve Spring Preload.

The load deflection characteristics of the piston rod were calculated using the formula for a cantilever beam with an intermediate load: (Reference 3)

$$y = -\frac{W}{6EI}(3a^2\ell - a^3) = -1.924 \times 10^{-4} W \text{ in.} \quad (5)$$

where: $a = 5.75$ in.

$\ell = 6.83$ in. (fixed end of rod to top of piston)

W = applied load, lb.

E and I same as previous values

The top of the piston will be deflected 0.010 in. radially by a side load of 52 lb. applied at the free end of the rod (approximately the bottom of the piston).

3.0 TASK II EXPANDER MATERIALS

3.1 Cost Data on Candidate Materials

Planning was initiated to assemble cost data on primary and alternate candidate materials for the corsshead and trunk piston design steam expanders components. Only preliminary data have been obtained to date in response to an attempt to determine the influence of alloy content on the price/lb. for bar stock, plate and castings.

From the preliminary data, the addition of approximately 1% Cr + 0.20% Mo (AISI 4140) to AISI 1045 steel represents an increase in price of the finished mill produce to about $0.053/lb. for quantities in excess of 10,000 pounds. Similarly an increase of an additional 1.0% Cr + 0.5% Mo to 1-1/4 Cr-1/2 Mo alloy steel plate increases the price by about $0.058/lb.

4.0 TASK III LUBRICATION

4.1 Recommend Liquid Lubricant for the Crankcase

The Mobil synthetic hydrocarbon oil designated XRN 1301-C has been selected for use in the first single cylinder steam expander test. This oil was specially formulated by Mobil for use in this program. The synthetic hydrocarbon base was selected because of its excellant thermal stability, temperature-viscosity characteristics and low pour point. Appropriate non-metallic inhibitors have been added to the base to provide the necessary oxidation resistance, rust protection, anti-wear properties and defoamant characteristics. Properties of the oil are shown in Table 4-1. In general, the XRN 1301-C oil should have approximately a 100°F advantage in use temperature (bulk oil temperature and

Table 4-1

Properties of Experimental Synthetic Hydrocarbon Lubricant[a]

XRN 1301-C

Gravity, °API	37.4	
Specific Gravity	0.8378	
Pour Point, °F	<-65	
Flash Point COC, °F	460	
Viscosity, cs @ 100°F	75.45	
SUS @ 100°F	349.7	
cs @ 210°F	11.07	
SUS @ 210°F	63.1	
Viscosity Index	147	
Acid No. ASTM D-664-1	0.06	
Base No. ASTM D-664-3	0	
ASTM Foam Seq. I Tendency, ml	350	
Stability, ml	0	
Seq. II Tendency, ml	40	
Stability, ml	0	
Seq. III Tendency, ml	410	
Stability, ml	0	
ASTM Rust, Distilled Water	Pass, Pass	
ASTM Rust, Sediment Water	Pass, Pass	
Emulsion ASTM D-1401		
Total Emulsion, ml	5	7
Time for 3 ml, min.	32	31
Time for complete break, min.	35	37

[a] Mobil Research and Development Corp.

hot spot temperature) over the oils based on the petroleum base stocks. Probably the greatest disadvantage of the XRN 1301-C oil over the natural hydrocarbon oils is its poorer demulsibility characteristics. However, it is believed the demulsibility of the XRN 1301-C is satisfactory for use in the steam expander. No existing commercial liquid lubricant was found to satisfy the crankcase lubrication requirements of the reciprocating steam expander without some modification of the additives.

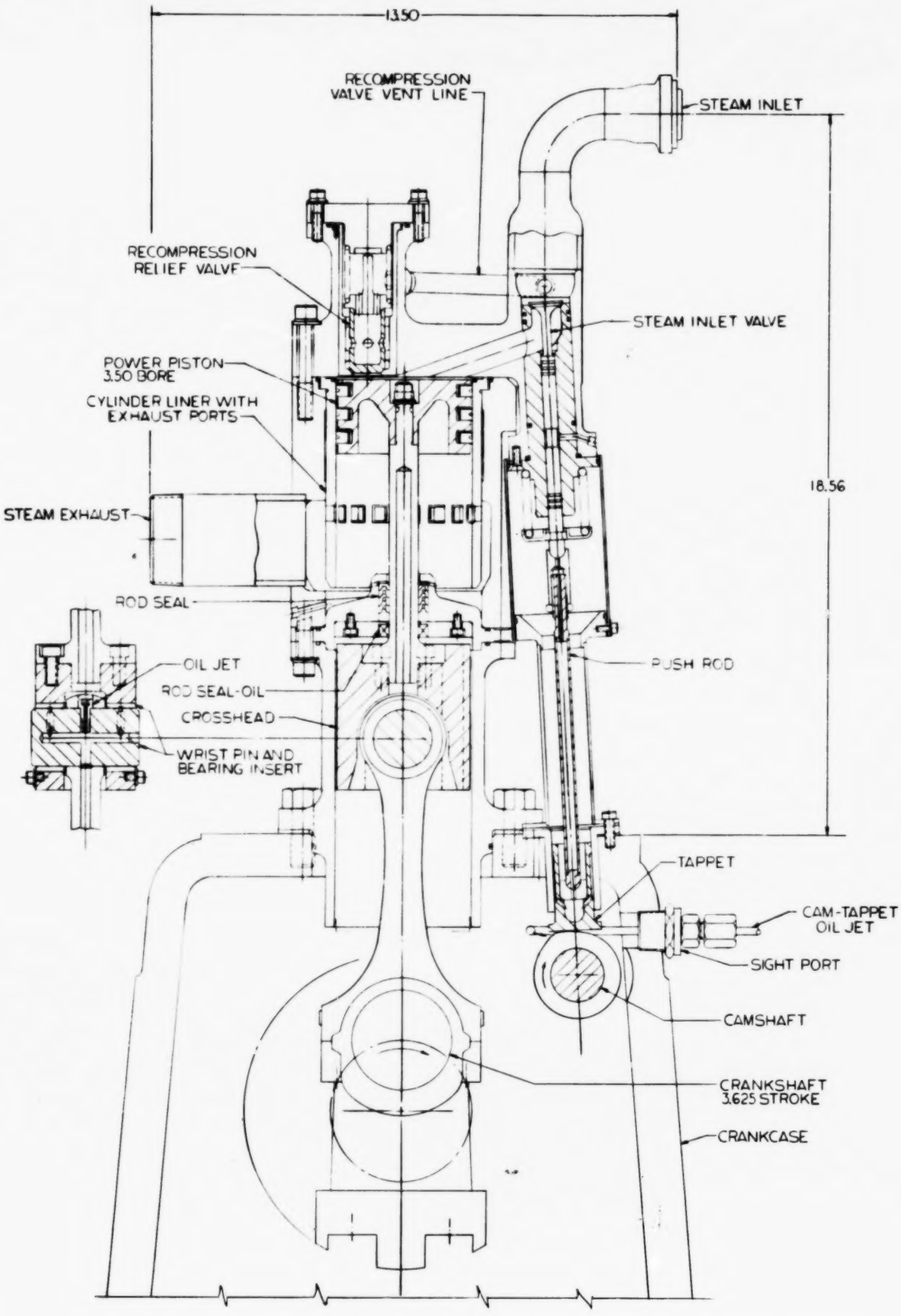

Figure 5.1-1. Assembly of Crosshead Expander (Dwg. 707E742).

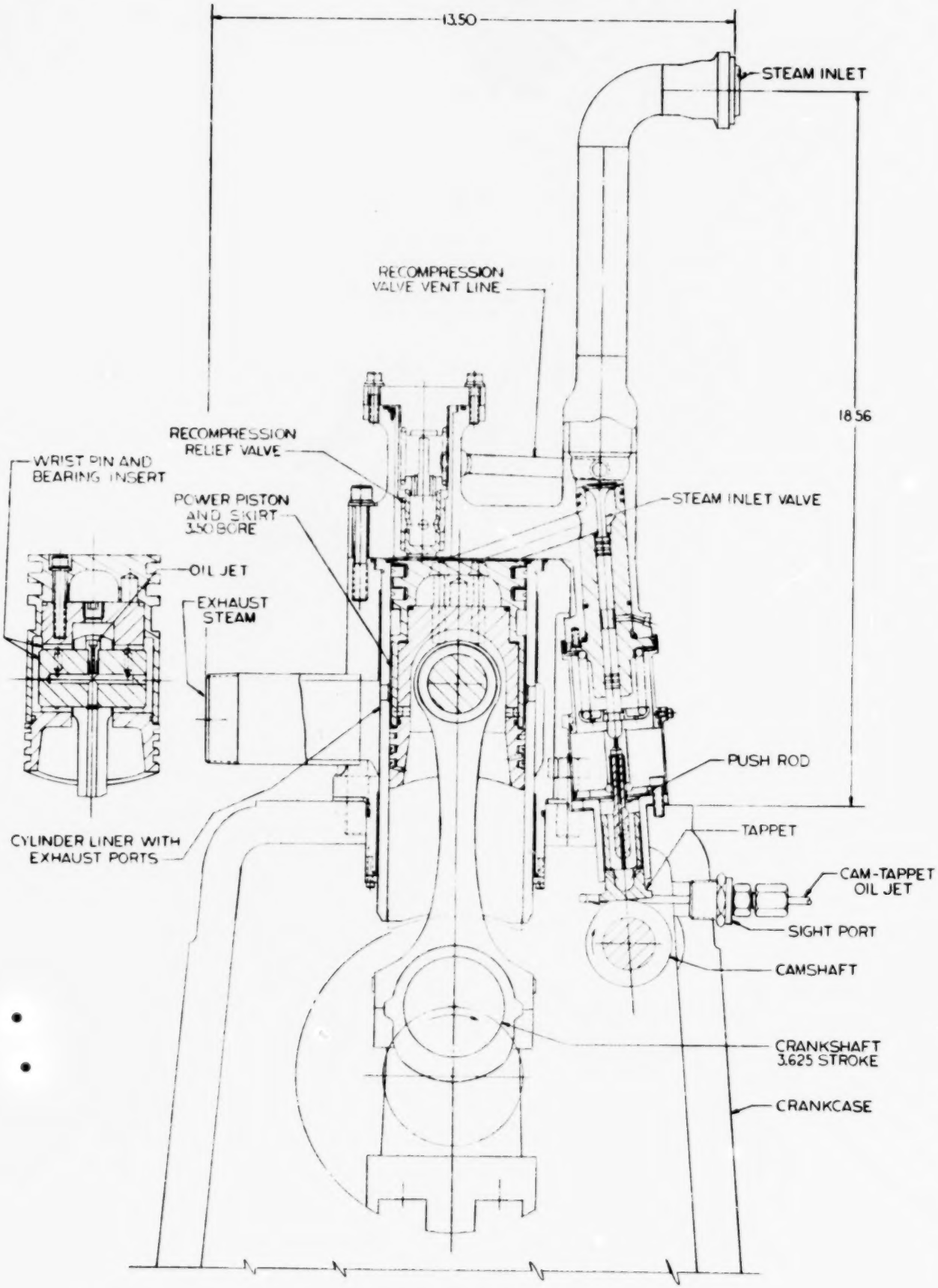

Figure 5.1-2. Assembly of Trunk Piston Expander (Dwg. 707E743).

Figure 5.1-3. Camshaft (P72-3-4S).

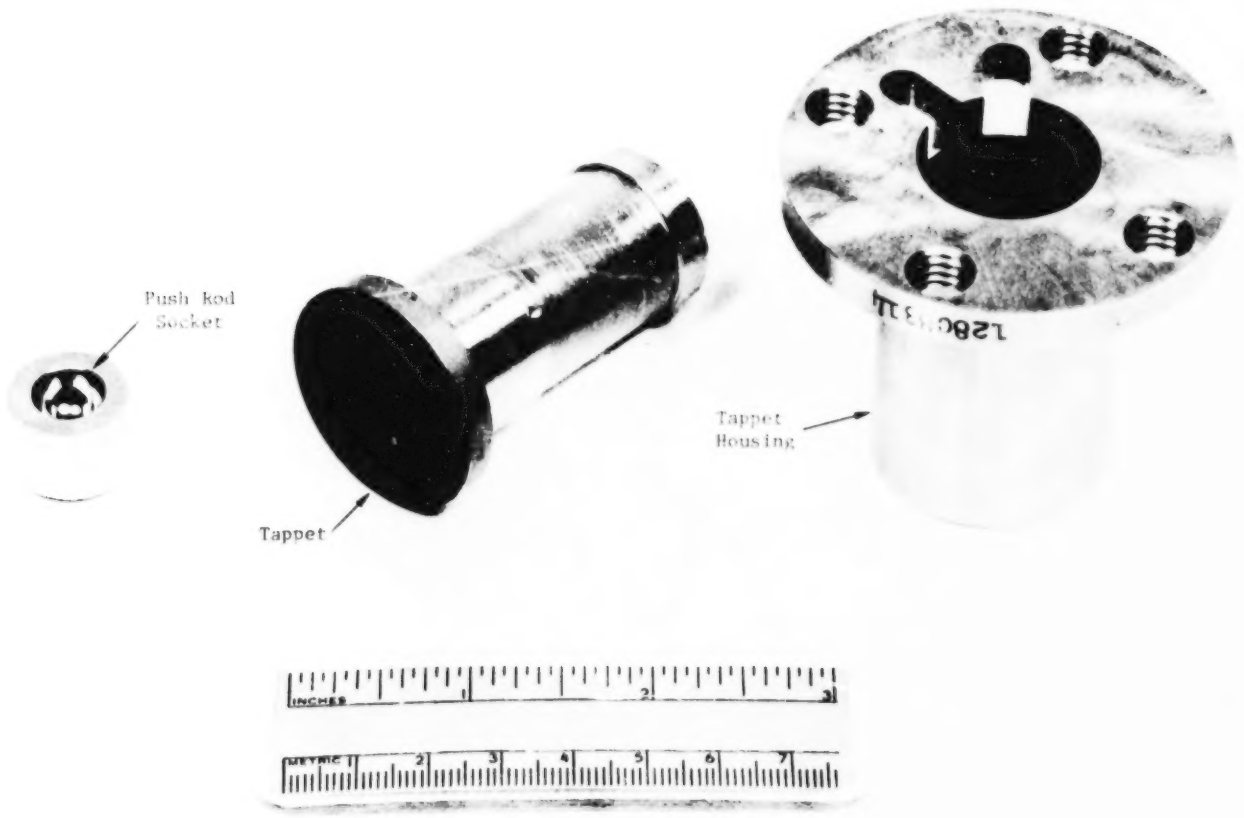

Figure 5.1-4. Cam Tappet Assembly (P72-3-4D).

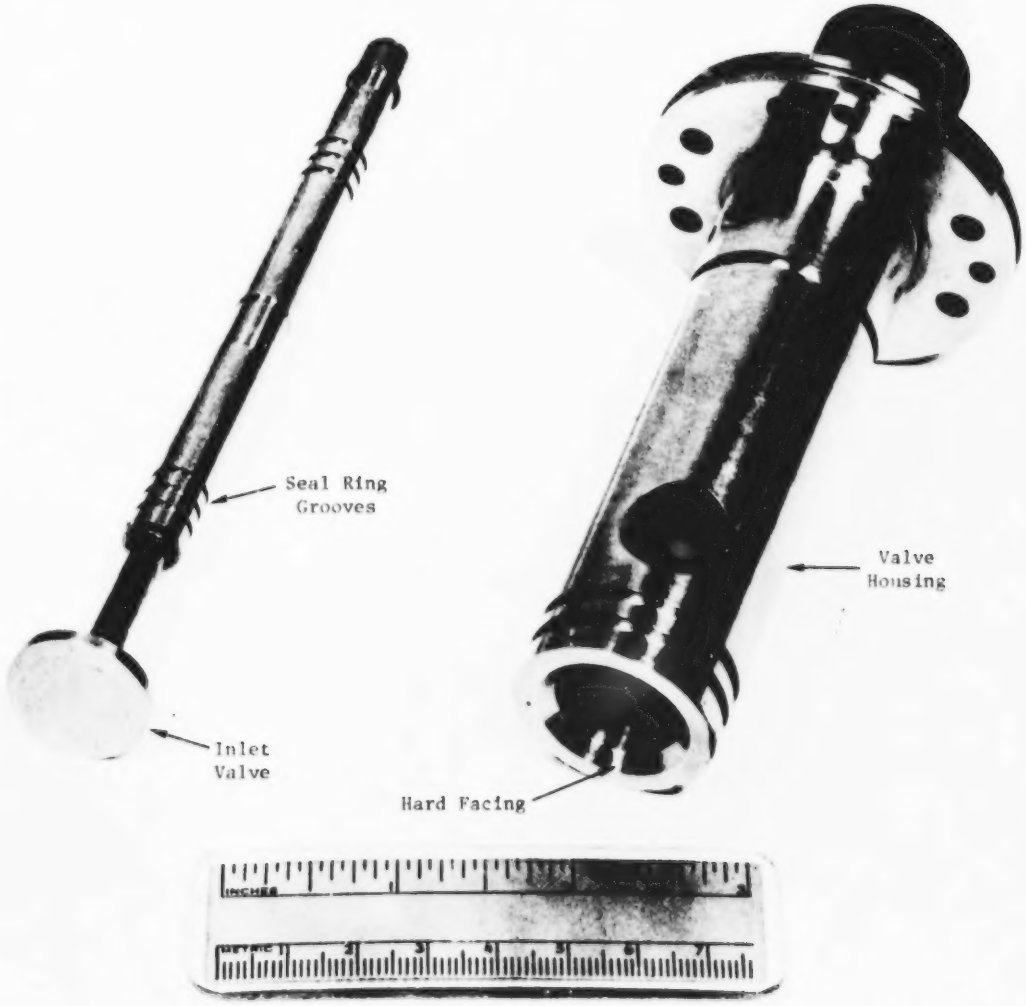

Figure 5.1-5. Steam Inlet Valve Assembly (P72-3-4F).

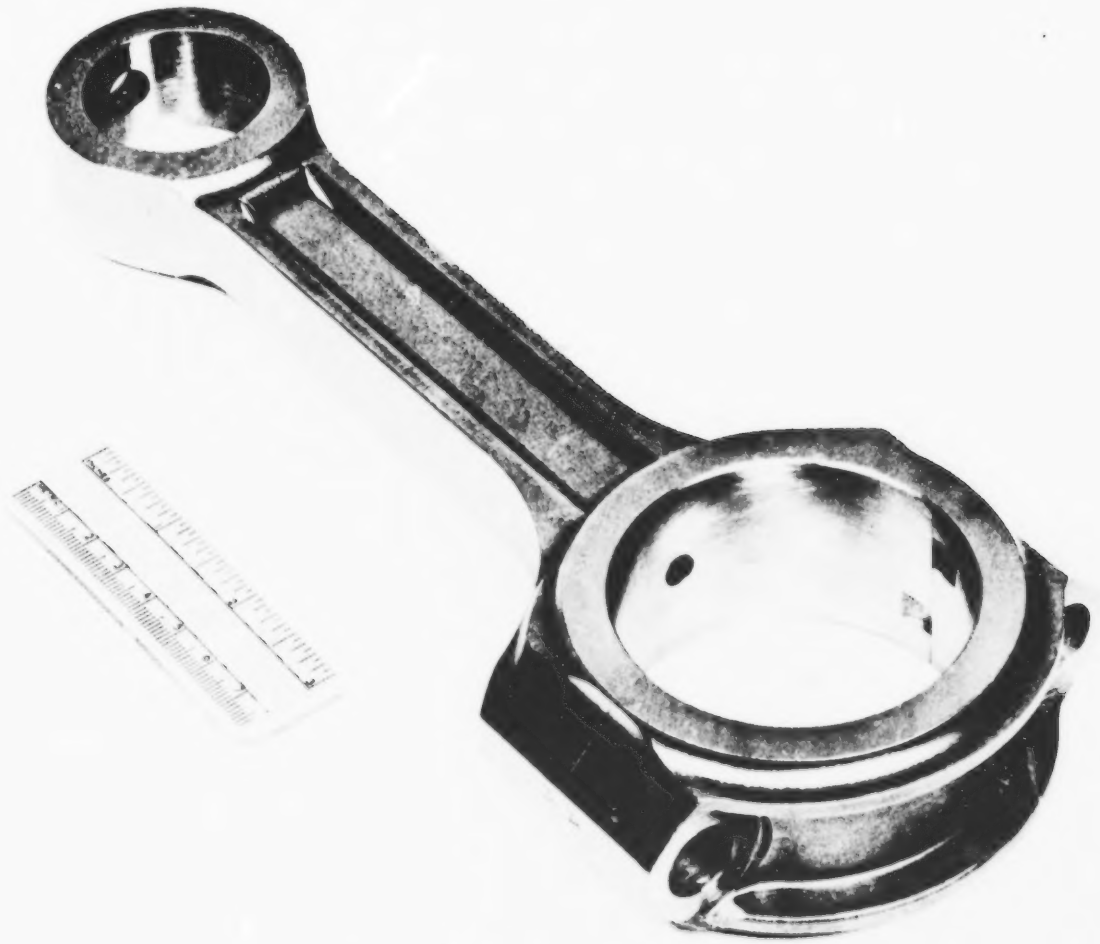

Figure 5.1-6. Connecting Rod (P72-3-4H).

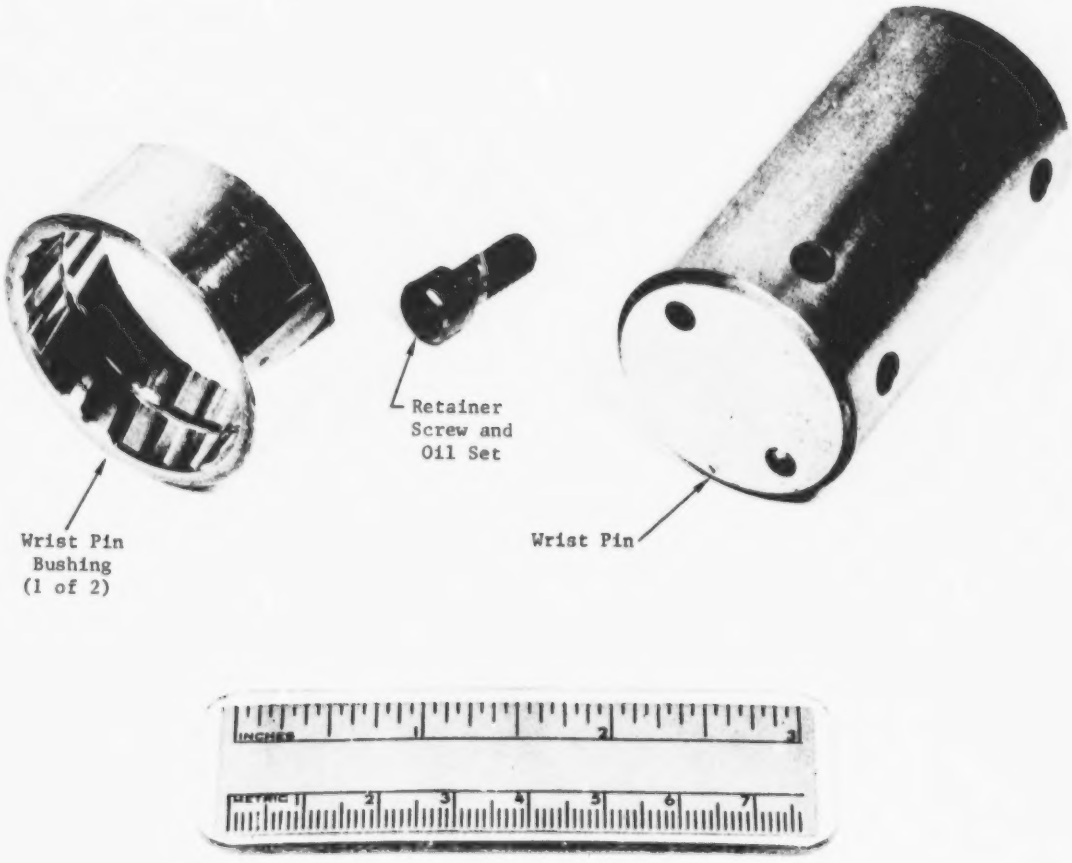

Figure 5.1-7. Wrist Pin and Bushing (P72-3-4G).

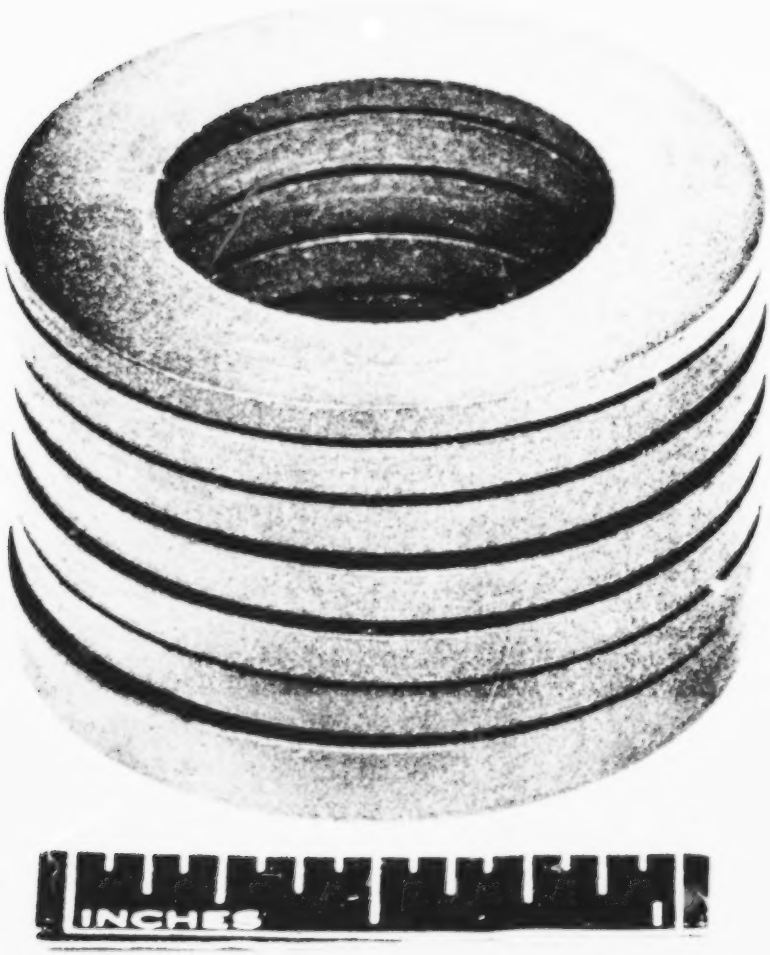

Figure 5.1-8. Power Push Rod Chevron Seal (P72-3-4M).

Figure 5.1-9. Power Push Rod (P72-3-4A).

Figure 5.1-10. Aluminum Crosshead (P72-3-4K).

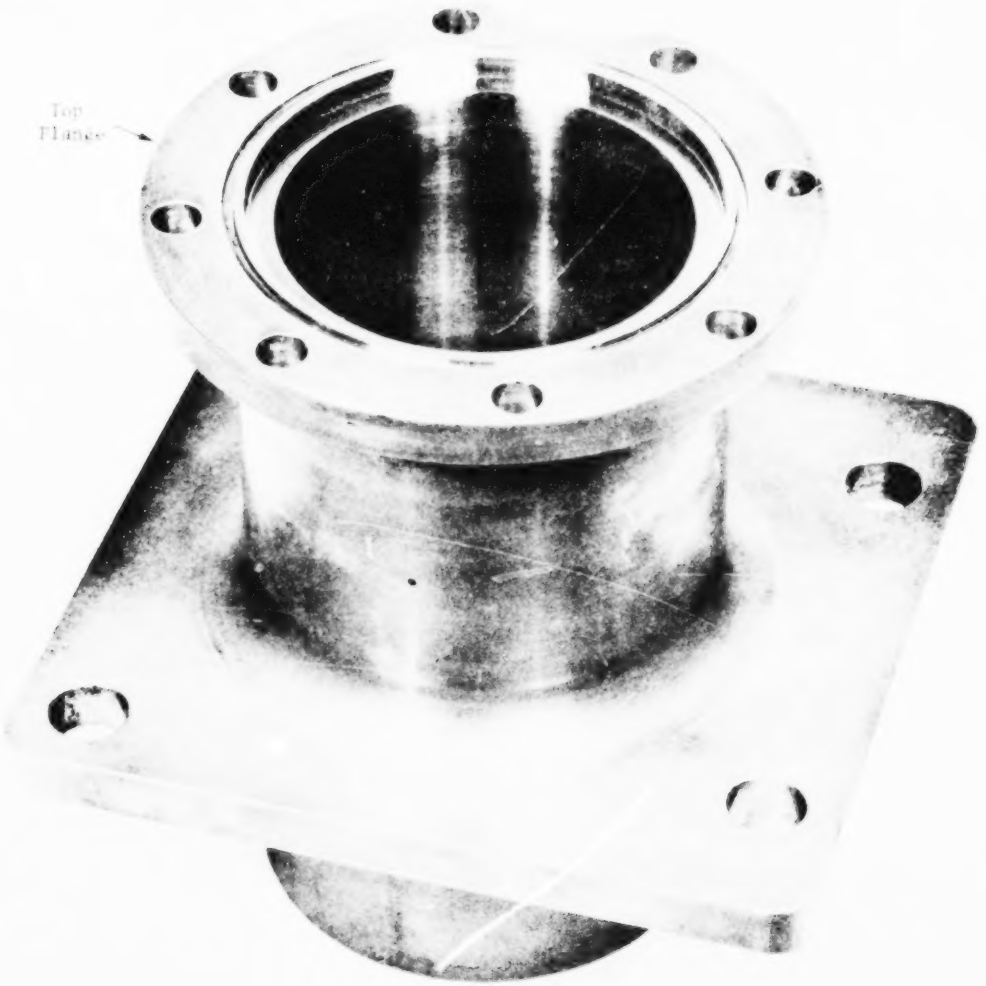

Figure 5.1-11. Crosshead Cylinder (P72-3-4J).

Figure 5.1-12. Power Piston for Carbon-Graphite Rings (P72-3-4U).

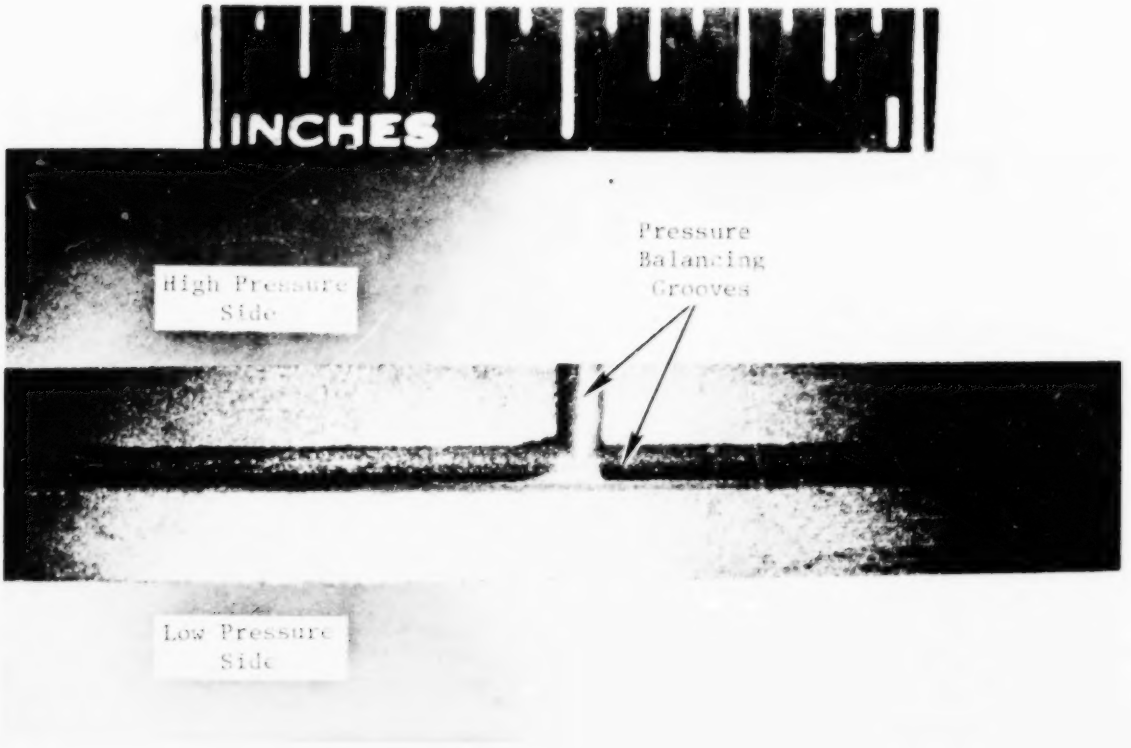

Figure 5.1-13. Pressure Balanced Carbon-Graphite Piston Ring (P72-3-2AD).

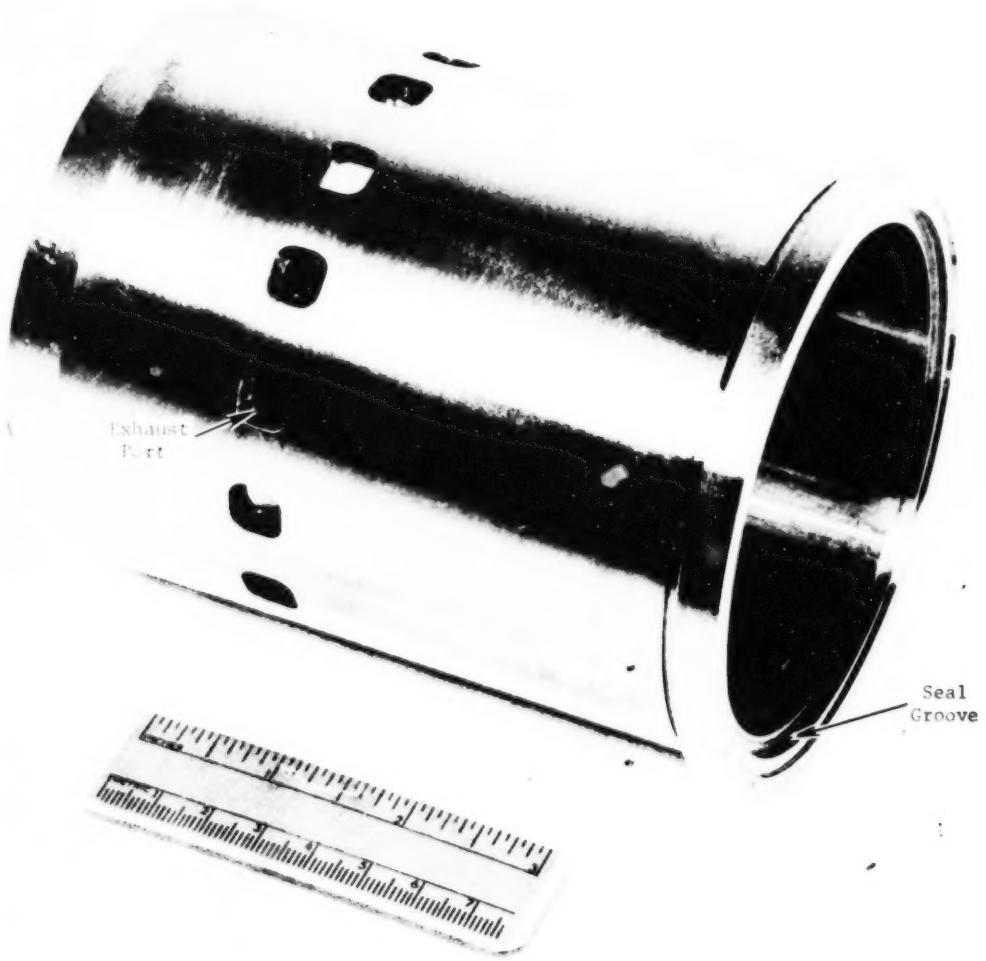

Figure 5.1-14. Cylinder Liner (P72-3-4C).

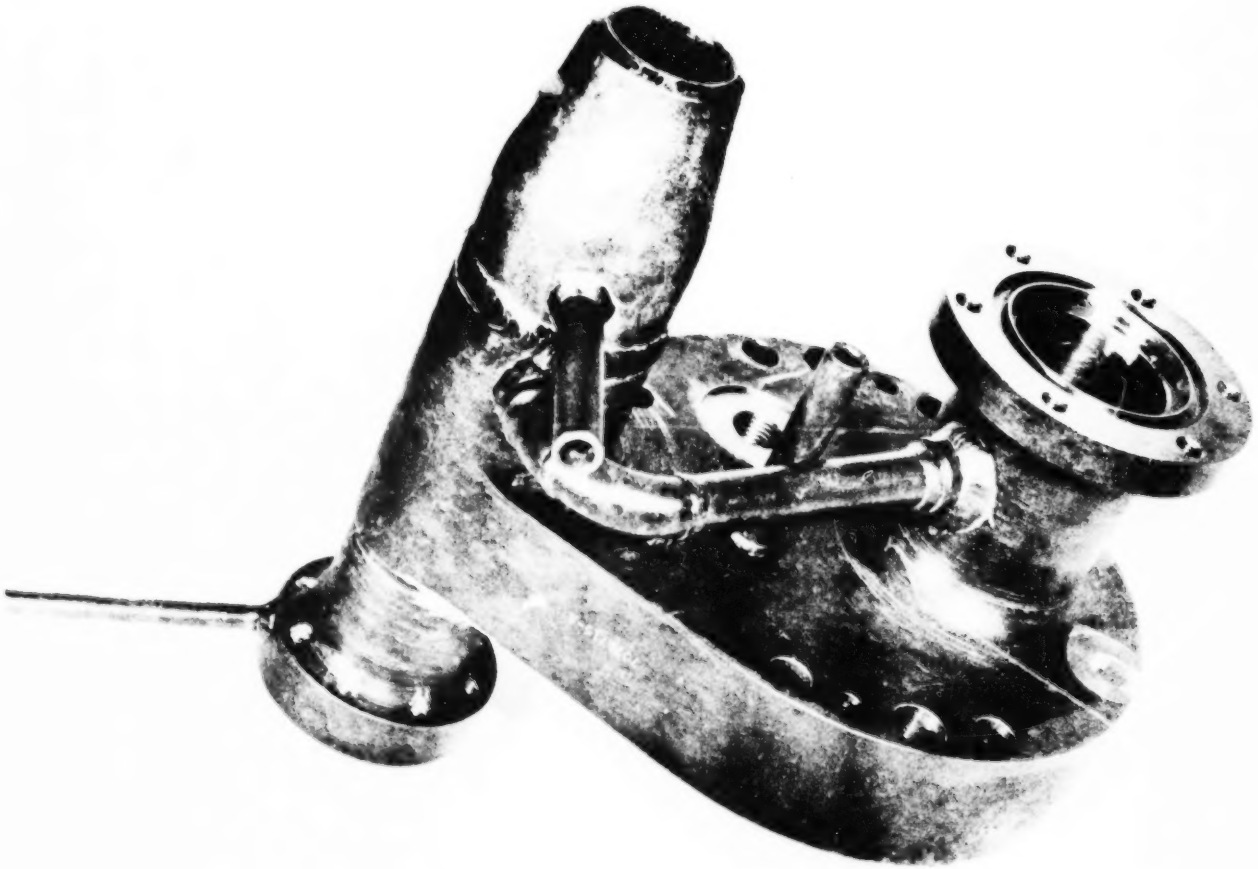

Figure 5.1-15. Cylinder Head (P72-3-4N).

to be solid lubricant or which are to function in the absence of liquid lubrication.

Buildup of the crosshead expander was performed in accordance with a written procedure which took into account the necessary dimensional checks to insure proper clearances and the necessary bolt torque valves. The camshaft was timed so that the steam inlet valve opened 10° B.T.D.C. The valve push rod was set with a cold clearance of 0.012 inch. This clearance was calculated using an earlier calculated temperature profile for the expander so that the hot clearance would be 0.005 to 0.008 inch.

Figure 5.1-16 shows a partial buildup of the Waukesha crankcase showing the required modifications to the gear train to provide one-to-one speed ratio between the crankshaft and camshaft. The expander was sent to the test cell on March 23rd for installation. Figures 5.1-17 and 5.1-18 show the crosshead expander in the test cell ready for initial motoring (driven by the dynamometer).

During this quarter concentrated effort has also been made to accumulate all of the trunk piston expander components. At the present time all parts are expected to be on hand for expander buildup during the later part of April.

5.2 Expander Testing

A detailed "Test Plan" for evaluating and comparing the two types of expanders was prepared. Each expander will be evaluated for its thermodynamic, mechanical, and material performance. A comparison of internal leakage (steam and oil) and frictional losses will be made. Emphasis will be placed on the durability of the solid lubricant components and the durability of those components which have no or limited liquid lubrication.

Figure 5.2-1 indicates the overall instrumentation to be used in expander testing. Figure 5.2-2 shows the detailed location of thermocouples and other items of instrumentation. Figures 5.1-17 and 5.1-18 show some of this instrumentation as it exists in the test cell.

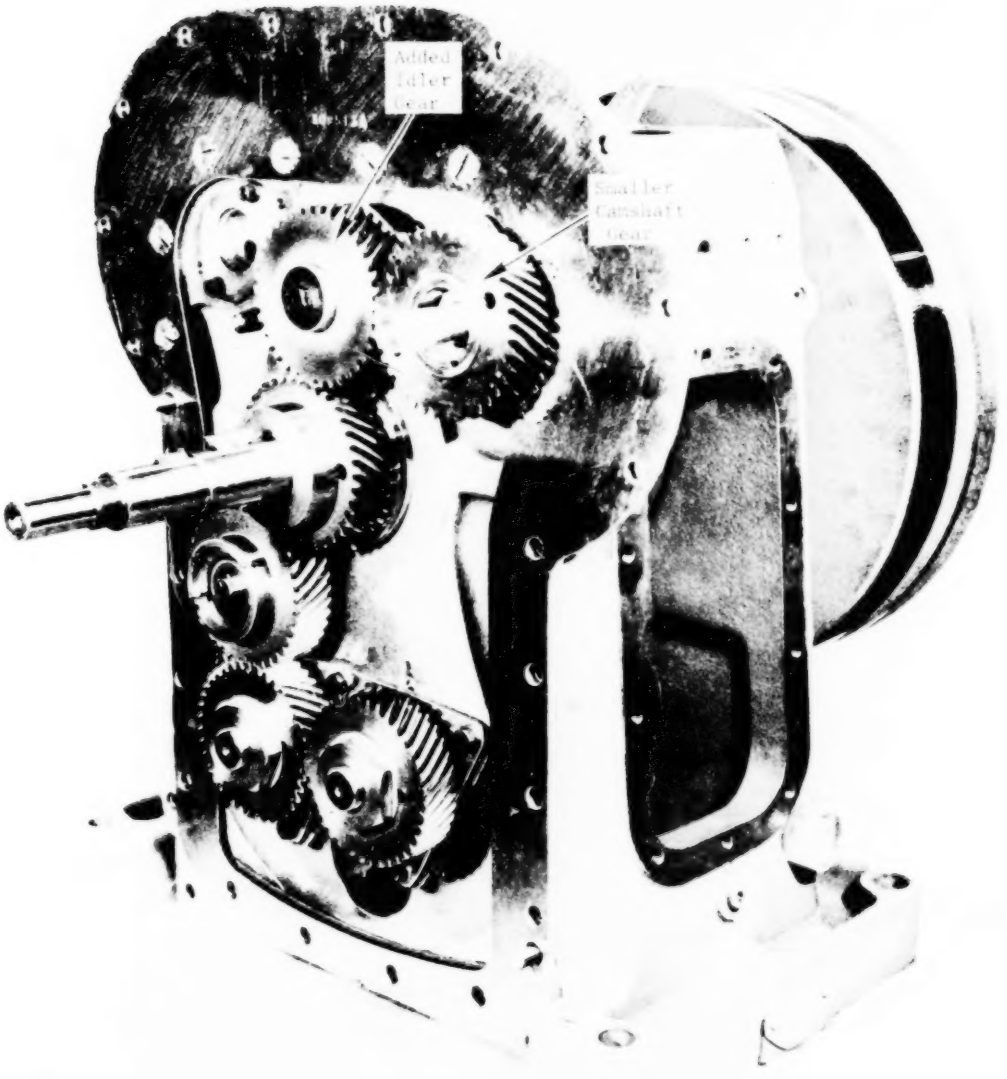

Figure 5.1-16. Waukesha Crankcase Showing Modified Gear Train (P72-2-5B).

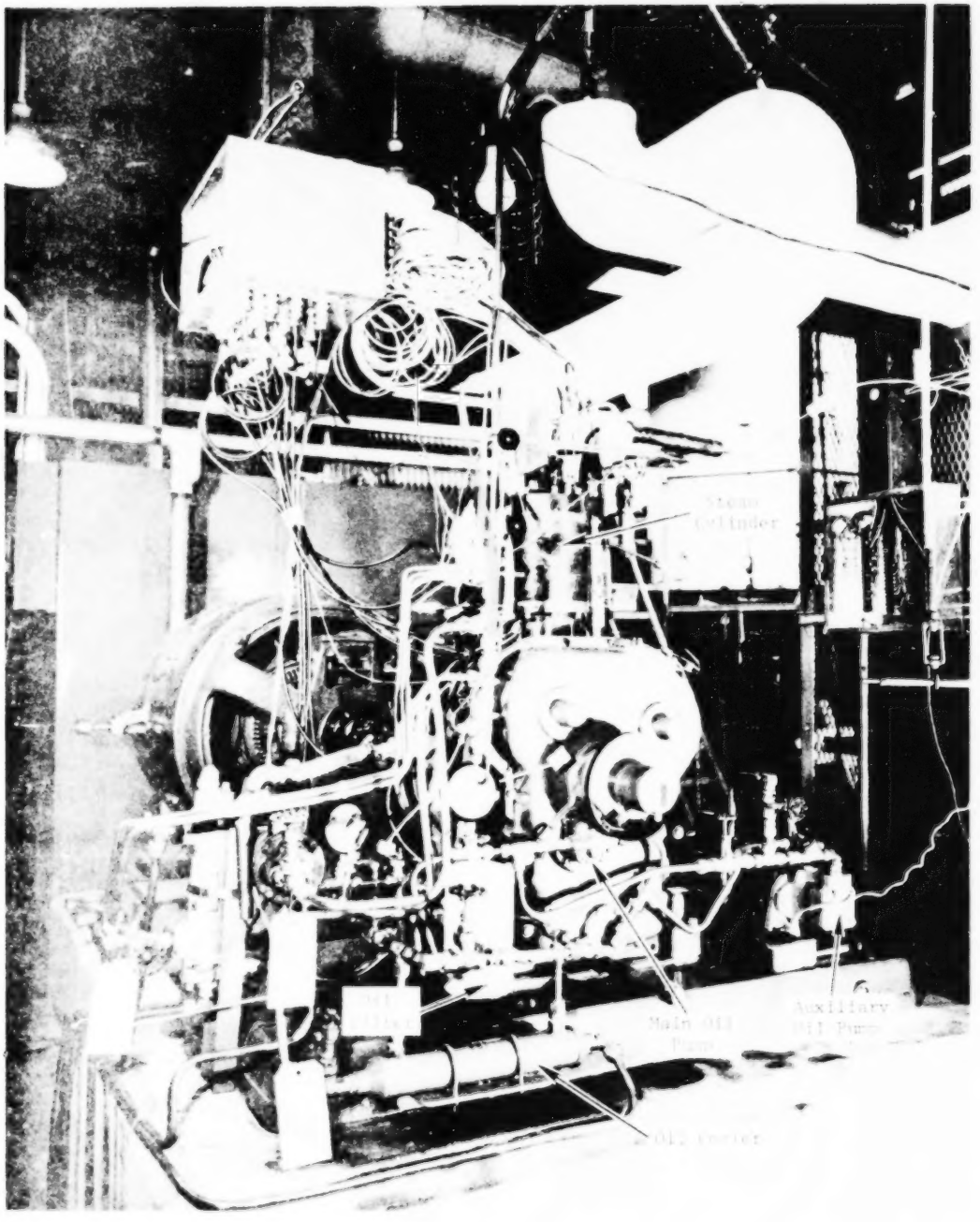

Figure 5.1-17. Installation of Crosshead Expander (P72-4-4N).

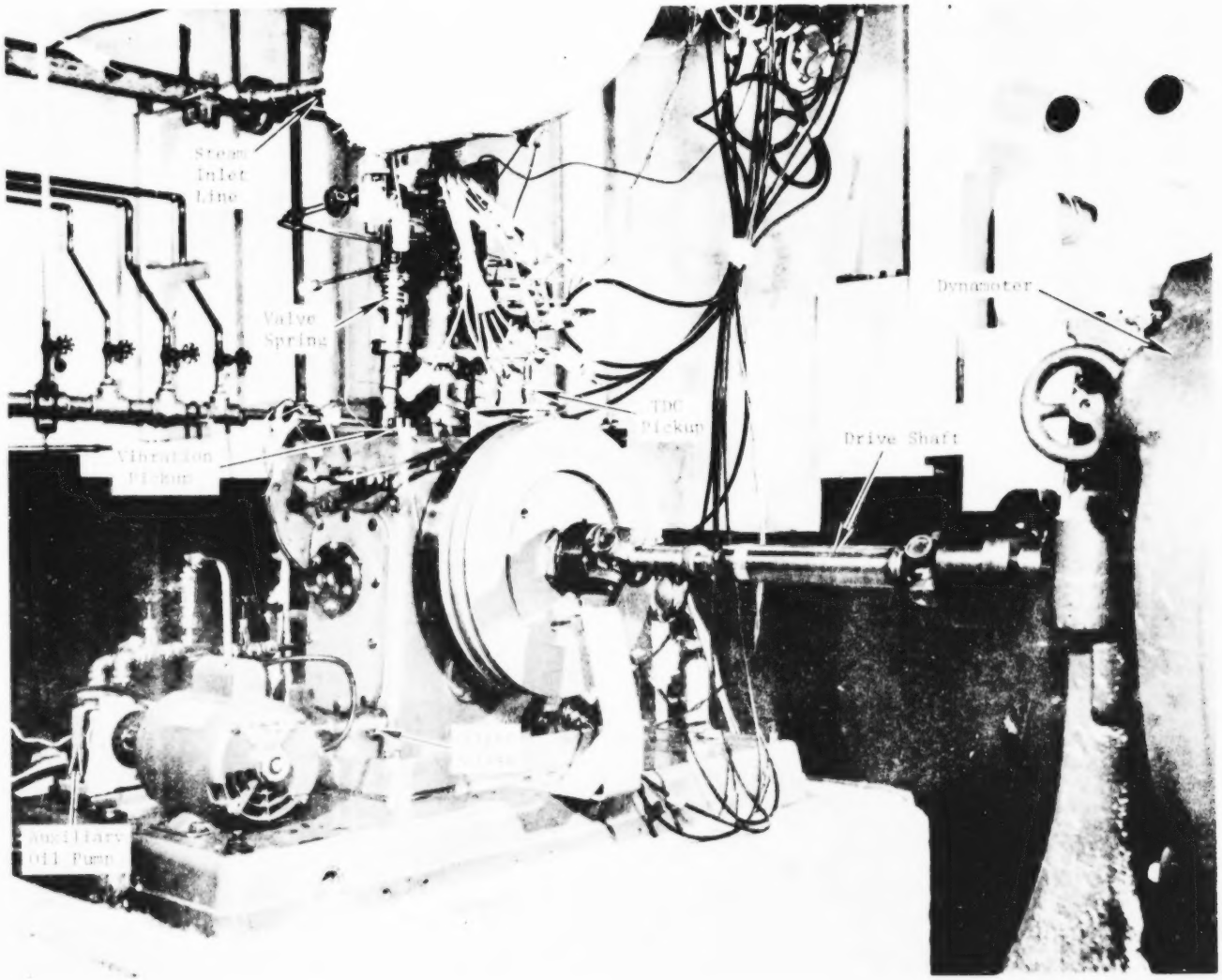

Figure 5.1-18. Installation of Crosshead Expander (P72-4-4M).

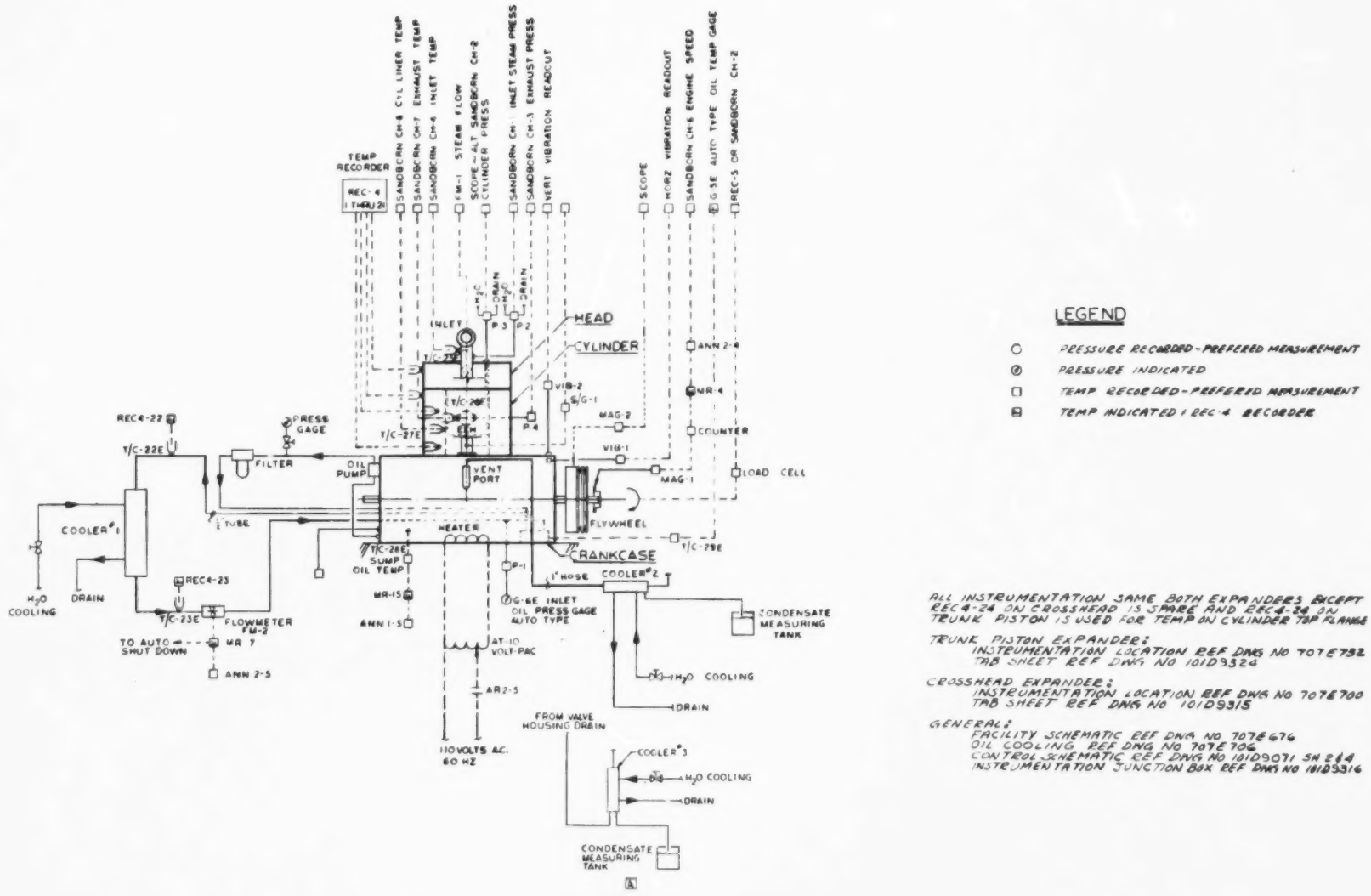

Figure 5.2-1. Instrumentation Schematic (Dwg. 101D9300).

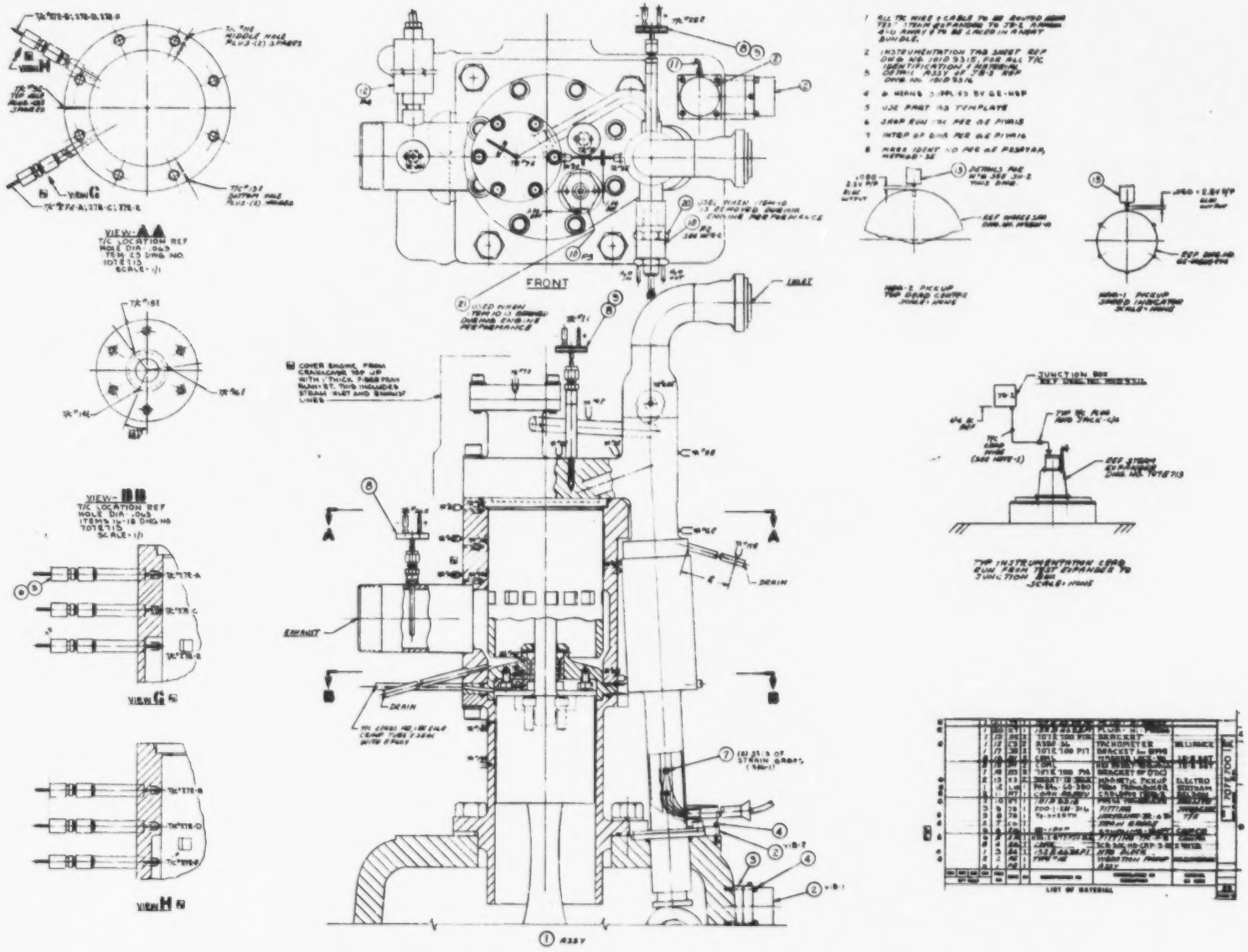

Figure 5.2-2. Thermocouple Location (Dwg. 707E700, Sh.1).

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So far no steam tests have been made with the crosshead expander. Rotation of the expander utilizing the dynamometer was first accomplished on March 29, 1972. The expander was rotated for about one hour at 500 RPM and for about one hour at 1000 RPM. Speed was increased to about 1200 RPM for a few minutes. All motoring tests were made with the recompression relief valve removed from the cylinder head. This was to minimize recompression pressure in cylinder. Significant results from this test are as follows:

1. The main oil pump which is part of the Waukesha crankcase could not maintain sufficient oil flow and oil pressure at low engine RPM. Oil pressure was 15 psig at 1000 RPM. An auxiliary oil pump is being installed to maintain 30 to 40 psig oil pressure to all speeds.

2. Additional oil preheating capacity is required to maintain 200°F oil sump temperature. The crankcase has a 1000 watt electric heater. Additional oil heating capacity is being installed.

3. Traces of oil entered the expander steam chamber within 20 minutes after the start of motoring through the power push rod seal. A change in the mounting of the seal to take advantage of hydrodynamic action of the seal has solved this problem.

4. Both the cam and the cam tappet were damaged during the two hour motoring test. This situation resulted in installation of a chilled iron tappet.

5. The power piston made light contact with the top of the cylinder liner. Running clearance between crosshead and its cylinder were too large at the low temperatures of motoring operation. Corrective action was taken to reduce the running clearance by slightly enlarging the diameter of the crosshead. Also the oil temperature to the aluminum crosshead will be increased in future motoring operation. Thermal expansion of aluminum will then further reduce clearance.

6. Expander vibration was quite low at 1000 RPM. Both horizontal and vertical vibration at the top of the crankcase was less than 0.1 mil.

7. Motoring power (no recompression) quite low at 1000 RPM - less than 1.6 hp required.

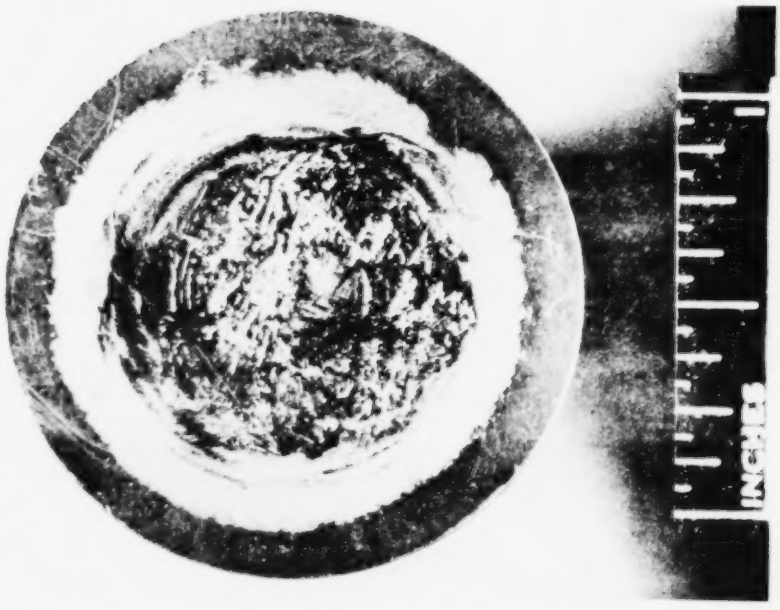
5.3 Steam Expander Test Facility

Installation of the test facility with its I^2R electrical steam generator has been completed. Initial checkout runs have been made using the expander by-pass steam flow control valve to control steam flowrate. Steam has been produced at 1000°F and 1000 psig at about 200 lb/hr. The desuperheated downstream of the by-pass throttling valve prevented full flow operation. This unit was intended only for operation with the expander in operation. It will be removed to permit full facility flow to the condensor to be tested prior to expander operation.

The steam expanders, especially the trunk piston expander, may allow significant amounts of lubricating oil to enter the test facility. Coking of oil over a period of time could cause crud deposits in the steam generator. Therefore, the facility is being modified so that it can be operated as an open loop system as well as a closed loop system. Operation of the facility under full rated conditions is expected to be accomplished the middle of April.

5.4 Evaluation of Tappet/Cam Materials

Upon disassembly from the engine, the surfaces of the cam and tappet were found to be disturbed. Visual examination of the surfaces at magnifications of 10X revealed the Stellite 6B tappet surface to show evidence of the classical appearance of spalling (pitting), i.e., irregular shaped holes apparently caused by rupture of the metal below the surface, and scuffing due to a galling action between the tappet and cam surfaces. The appearance of the surface of the tappet can be seen in Figure 5.4-1. Visual examination of the carburized AISI 8620 steel cam surface revealed a buildup of metal on the nose and on each side of the lobe of the cam surface, Figure 5.4-2. The metal transfer to the cam surface is believed to be caused by a galling action between the tappet and cam surface as a



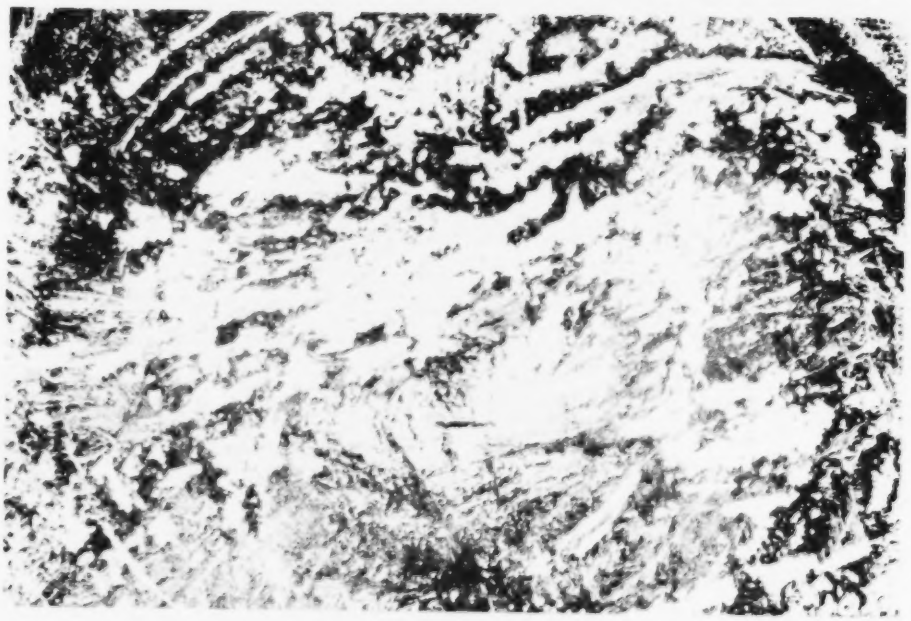


Figure 5.4-1. Photograph of Stellite 6B Tappet Surface Showing Spalling and Possibly Scuffing Due to a Galling Action Between the Stellite 6B and Carburized AISI 8620 Steel Cam Surface. Top 3X, Bottom 10X. (P72-4-1A, P72-4-1B).

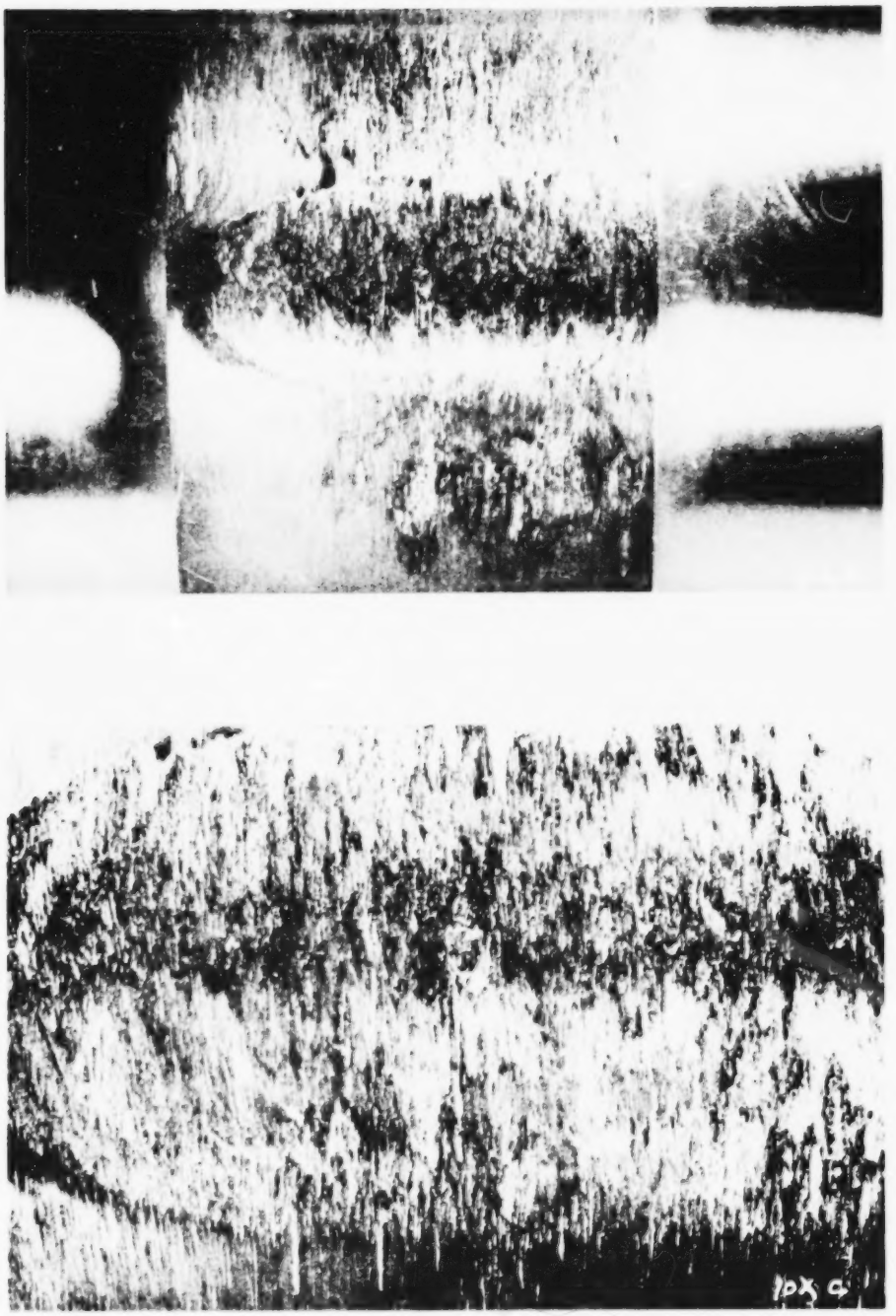


Figure 5.4-2. Photograph of Carburized AISI 8620 Steel Cam Surface **Showing** Metal Transfer from Tappet. Top 4X, Bottom 10X. (P72-4-1E, P72-4-1F).

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result of a combination of a high tendency for adhesion between the two surfaces and a low bulk fracture strength of the Stellite 6B. Initiation of this action could have been the result of lack of lubrication during startup of the engine prior to buildup of sufficient oil pressure.

The selection of Stellite 6B (1.2% C - 30% Cr - 4.5% W-Bal Co) for the tappet was made on the basis of its availability in a wrought form and its generally good wear resistant characteristics dur to the relatively high carbide content in a Co-Cr-W matrix. However, in discussions with representatives from Ford Motor Co., Eaton Manufacturing Co., Waukesha Motor Co. and Mobil Research and Development Corp., it was concluded without question that the bulk hardness (strength) of Stellite 6B (Rc 38-39) was too soft for application as a cam follower. Of the Stellites, it is possible that the Star J casting alloy (2.5% C - 32.5% Cr - 17.5% W - Bal Co) with a chill-cast hardness of Rc 61 may have worked satisfactorily against the carburized AISI 8620 steel cam surface. However, experience has shown that the Star J alloy shows superior performance against a nitrogen rich surface. It should be noted that if a nitrided surface is to be used for the cam, a carbonitride process should be utilized in order to obtain the desired case depth to carry the high loads. The core depth of a straight nitrided steel surface is too thin for application on cam or tappet surfaces.

The following material combinations were recommended as candidates for the cam and tappet in the second engine buildup:

	Cam	Tappet
1.	AISI 8620/carburized (Rc 58 min.)	Chilled Cast Iron (Rc 54) /Ferrox Coating
2.	AISI 8620/carburized (Rc 58 min.)	Carboloy 883 (WC + 6% Co) (R_A 92)
3.	AISI 8620/carbonitrided (Rc 60-65)	Star J (Rc 61)
4.	Hardenable Cast Iron (Rc 54 min.)	Hardenable Cast Iron (Rc 55-60) / Mn-Fe Phosphate Coating
5.	Hardenable Cast Iron (Rc 54 min.)	D-2 Tool Steel (Rc 58-60)/Mn-Fe Phosphate Coating

For the obvious reason of availability, the existing carburized AISI 8620 steel camshaft was selected for the second engine buildup. For the tappet material, the conventional steam tempered (black oxide coating)

chilled cast iron was recommended. Experience has shown that chilled cast iron is generally superior to hardenable cast iron against hardened steel cam furfaces[5]. The major disadvantage of chilled cast iron tappets is its greater tendency to failure by spalling (pitting) expecially in the presence of certain EP additives in the oil[6,7,8,9]. However, EP additives in the oil are needed for wear and scuff resistance of the chilled cast iron.

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